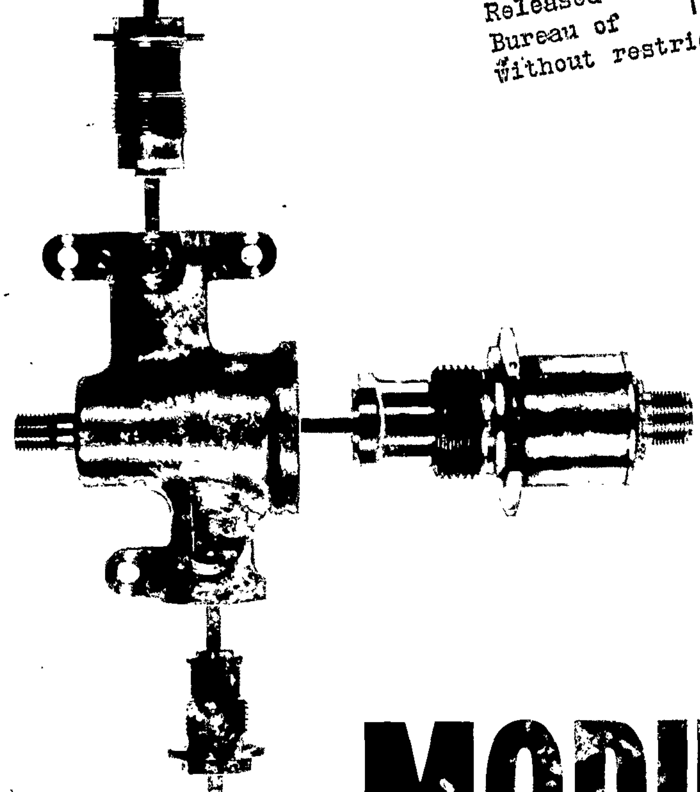

MODULAR HYDRAULICS

FINAL REPORT

BuWeps Contract 59-6019 C

PART I

CHANCE VOUGHT AERONAUTICS DIVISION

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MODULAR HYDRAULIC SYSTEM DEVELOPMENT
PROJECT HYDRATOY

PART I
FINAL ENGINEERING REPORT
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ABSTRACT

This is the final report of a research and development program sponsored by the Airborne Equipment Division of the Bureau of Naval Weapons for the development of Modular Hydraulic components and concepts. The program is informally called "Project Hydratoy" and was initiated in December of 1958. The technical monitor for the program was Mr. B. L. Mentee of the Airborne Equipment Division.

In general terms, the program's basic objectives are to:

1. Package groups of individual components into one housing to save weight and space and to gain reliability.

2. Make the use of packages more attractive and to simplify installation and maintenance by providing a standard line of self-contained cartridge-like components for use in these packages.

3. Carry this integration one step further and investigate ways and means of physically integrating the complete hydraulic system into its supporting structure.

4. Develop the above concepts for a 450°F, 4,000 psi hydraulic system using corrosion-resistant materials and metallic seals.

This report is published in four separately bound parts:

Part I contains results of metallic seal and package development.

Part II presents results of modular component development and the specifications and standards for those components.

Part III contains results of development in the integrated system concept and design criteria for that concept.

Part IV is a report of materials and process development which occurred in conjunction with and as a result of development effort in metallic seals, packages, components, and system integration.

NOTICE

ACKNOWLEDGEMENTS

The following persons have contributed to the metallic seal and package development work reported herein:

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SECTION I

INTRODUCTION

A. Introduction

This is the final report of a research and development program sponsored by the Airborne Equipment Section of the Bureau of Naval Weapons for the development of Modular Hydraulic components and concepts. This program, informally called "Project Hydratoy," was initiated in January, 1959, with coordination conferences with the services, various airframe contractors, and the vendor industry.

In general terms, the program's basic objectives are to:

1. Package groups of individual components into one housing to save weight and space and to gain reliability.

2. Make the use of packages more attractive and simplify installation and maintenance by providing a standard line of self-contained cartridge-like components for use in these packages.

3. Carry this integration one step further and investigate ways and means of physically integrating the complete hydraulic system into its supporting structure.

4. Develop the above concepts for a 450°F, 4,000 psi system using corrosion-resistant materials and metallic seals.

B. Definitions

Before proceeding further, several definitions should be established. The terms <u>modular</u>, <u>module</u>, and <u>package</u> will be used frequently throughout the report, so an understanding of the meaning of these terms is in order.

The word <u>module</u> means a model or an ideal shape. This term is used to identify the standard cartridge components. <u>Modular</u>, which means to proportion or arrange, is used to describe the design arrangement or integration of the modules in a system design. A <u>package</u>, as used here, describes a common housing for two or more components which contain all internal passageways necessary for component function. Modular design in architecture or electronics is used with the goals in mind of utmost efficiency and simplicity. This program's goal in mind is the design of a hydraulic system

based on the use of standard modules whose various parts are arranged to produce a high degree of simplicity and whose system network with other modules is integrated with the basic frame to increase the efficiency of the vehicle as a whole.

C. **Need**

In the present-day aircraft, hydraulic systems consist of individual components located throughout the aircraft with a large number of interconnecting tubes. This type of hydraulic system constitutes one of the major maintenance problem areas, especially in fleet operations. The over-all reliability of present systems is low. In addition, the need for individually tailoring the external configuration and attachments for components in high density aircraft has resulted in a drastic deterioration of standardization. This lack of standardization has also created logistic problems. This program has attempted to renew standardization of components and present a new system design approach which will allow a substantial increase in the reliability potential of a system.

The high performance systems needed for this and the coming age require a high degree of reliability with a minimum of maintenance, and it will not be possible to build these factors in "after the fact" as has been partially done in the past with constant revision and modification to systems that were installed after the vehicle structure was completed.

Improvement of any fluid system in regard to reliability and maintenance will require several things:

1. An investigation of the fluid system design and installation characteristics to produce an analysis of what needs to be done and why; and, in addition, a hefty slice of creative and ingenious ideas for new and better ways to improve the complete airframe systems integration.

2. An education program for designers and industry management to explain the problems and the objectives and show why the success of the integrated systems design concept is completely dependent upon starting the integration of the fluid system and the airframe with the first and initial designs.

What is meant by the term "integrated systems design"? As an example, consider the human body and its blood circulation system. From the power source, the heart, blood is pumped to all parts of the body through a distribution system consisting of main trunk lines (the arteries and veins) and secondary distribution lines. The main trunk lines are located adjacent to and parallel with the structural skeleton in such a manner that the maximum protection from external damage is achieved. Although we possibly cannot duplicate the self-healing characteristics of the body, repairs to a fluid system

integrated into the body or an airframe structure can still be accomplished although certainly not as readily as existing systems, but the probability of damage is a great deal less and, thus, increases the system's inherent reliability. An optimum visualization of a completely integrated fluid systems design would picture a system whose transmission lines or flow passageways are an integral part of the structure or are permanently attached to the structure so that the maximum protection from external damage is achieved. The reliability would then be based on the reliability of the complete structure. The components of the system would be grouped into sub-system packages fed by the main trunk lines and would be placed at accessible points in the structure and near the end item they control. The tie-in to the main trunk line at the manifold would permit removal or installation of the package without the necessity of losing system fluid nor having to replace fluid after reinstallation. Use of the modular package would enable the components to be thoroughly checked and serviced at a test bench before installation, and the only leak points in the systems which would require maintenance surveillance and be involved in maintenance reliability would be the joint between the package and the permanent system.

The specific approach taken in this program was primarily concerned with the development of a series of standardized internal working parts, called modules, for all the types of components required in hydraulic system design (check valves, relief valves, shuttle valves, etc.). The airframe designer can use these standardized modules to build up systems of sub-systems required by his aircraft or missile. A built-up system would consist of a manifold containing the working-part modules required by the particular system, interconnected with the necessary fluid flow passages. It can be an integral package, the external configuration of which can be tailored to fit the specific installation. Or, the manifold may consist of structural parts used for other purposes. For example, the hydraulic sub-system used to actuate landing gear might be packaged in the mechanical linkage of the actuating system. In any event, with such concept all aircraft would use standard modules, the only difference between aircraft would be the way they are used which has no effect on maintenance and supply. In addition, a tremendous amount of simplification may also result since it may be found practicable to discontinue the present practice of having different size components for all different tubing sizes. Instead, cartridge components are sized on flow capacities.

D. Design Considerations

1. Design Areas

There are primarily four areas of design which require detail considerations of their effect on reliability:

a. System function design
b. Component design

c. Installation design
d. System connection design

Part of the philosophy of the modular concept is that these four areas must be individually and simultaneously considered before any one is brought to the final stage.

2. Criteria

To apply a concept for improving any of the four areas of a fluid system, there should be established a set of values by which specific solutions can be evaluated. Early in the modular program the following criteria were established to be used in all phases of the program in evaluating design approaches:

a. Increase the RELIABILITY of the complete system
b. Improve the utilization of SPACE required for the complete system
c. Increase the DESIGN FLEXIBILITY for systems during both initial design and for future changes to systems
d. Seek improved MAINTAINABILITY of the entire system and better service life with less system attention required
e. Reduce the WEIGHT of the complete system
f. Reduce the COST of the complete system during initial installation and during its service life

In certain designs, these factors may overlap other factors; for example, weight saving may result in a cost increase. Also, in certain applications in missiles, weight and space might be so critical that it affects all considerations. In general, an "order of importance" should be assigned to each factor, then each weighed separately when comparing optional designs or simultaneously when evaluating one design.

3. Evaluation

When a new design idea or concept is visualized, the first evaluation to be considered is what advantages does this new concept offer over the latest refinement of the concept to be replaced. Often when some new, interesting idea is conceived, we become intrigued with its freshness and challenge, and fail to make a general analysis of its over-all comparison to existing concepts for predicted application.

In preparing the analysis for a comparison of the new concept against the old, there are two premises that have to be made.

a. Has the old concept been used properly and all its potential realized? In other words, has the old concept been developed to the point of practical perfection. If not, it should be assumed for analysis comparisons that the old concept is perfected.

b. It should also be assumed when a comparison for a particular application is made that the new concept is also perfected, so that only its inherent general values (good or bad) are considered. This prevents the basic analysis from becoming stymied in detail design problems which need not be considered until the general analysis indicates that the basic idea involved is advantageous.

The various concepts of the modular approach were analyzed by the preceding criteria. However, in most instances, particularly the program efforts to mate the system and the airframe, the practice prevalent in the industry of installing the hydraulic system where there was room to squeeze it in without interferring with structure or other equipment, made a logical comparison of design fundamentals extremely difficult.

4. Design Use

The modular designs produced by this program were approached on the consideration that the eventual use would be in aerospace vehicles in production quantities. These designs were evaluated on the same basis but several premises or philosophies of use had to be assumed. These premises are:

a. Sufficient lead time would be available for package design and system installation design during basic design.

b. Simulated system tests would be performed before cast or forged packages are committed.

c. Installation or replacement of individual modules will be performed only at an established assembly station. Vehicle installation or replacement will be for complete packages only.

d. The use of a permanent type fluid connection (brazed fitting or equivalent) would be used throughout the system.

E. Objectives and Requirements

1. Objectives for Project Hydratoy

a. Develop a method that will permit renewed standardization of hydraulic components.

b. Establish and develop a line of standard Type III modular hydraulic system components.

c. Establish and develop methods for packaging modular components into complete hydraulic sub-systems.

d. Develop and test static type, all-metal seals to satisfy all requirements for sealing the modular components when installed in manifold cavities.

e. Develop methods for creating basic aircraft structural configurations which can do double duty by providing fluid power transmission paths in addition to performing the structural functions.

f. Develop methods for reducing or eliminating the quantities of hydraulic rigid and flexible lines and their attendant brackets, clamps, screws and clearance problems.

g. Develop methods for creating dynamic joints which can replace flexible hoses or tubing coils.

h. Evaluate the over-all feasibility of these new approaches to creating aircraft, missile, and space vehicle hydraulic systems.

2. Requirements

The basic general requirements for Project Hydratoy were established as follows:

a. System Fluid

MLO-8200 hydraulic fluid

b. System Pressures

4,000 psi -- normal operating pressure
6,000 psi at +450°F - component proof pressure
8,000 psi at +450°F - tubing and fitting proof pressure
10,000 psi at room temperature - component burst pressure
16,000 psi at room temperature - tubing and fittings burst pressure

c. Flow Rates -- 0 to 25 GPM (in three flow classes)

Class A --- 0 to 4 GPM
Class B --- 4 to 12 GPM
Class C --- 12 to 25 GPM

d. System Temperatures

Fluid Operating. . . -65°F to +450°F

Note: Components shall be capable of full performance with the fluid at any temperature throughout the range of -20°F to +450°F and shall start to operate at a temperature of -65°F.

Ambient . . . . -65°F to +650°F

Note: Solenoid-operated components shall be capable of operating within the range of ambient temperatures of -65°F to +650°F.

e. Seals -- All seals shall be metallic.

f. Materials -- Corrosion-resistant materials shall be specified.

F. Development Discussion

The program was divided into several concurrent phases. A summary discussion of each of these phases follows.

1. Module Development

The development of the various types of modular cartridge components consisted of three major steps:

a. Determination of unit envelopes and preparation of a detailed procurement specification.

b. Receipt and evaluation of competitive proposals and selection of vendors.

c. Development and qualification by selected vendors.

In the determination of the envelopes, several ground rules were established on the basis of program criteria and principles of hydraulic design. These were:

a. Module units shall be self-contained units such that a module may be removed from one housing and inserted into another without any detail disassembly, readjustment of settings or impairment of function.

b. The unit shall be designed to eliminate any possibility of the module being installed backwards.

c. Component size - Size of the miniature module components will not be based on tube sizes but will be divided generally into three classes according to flow requirements.

(1) 0 to 4 GPM
(2) 4 to 12 GPM
(3) 12 to 25 GPM

d. No unit of a particular size or type shall be interchangeable with any other type or size unit.

e. Units shall be designed for use normally with a face-type static metallic seal. (There is an exception to this rule.)

Based on the above rules, and on information willingly supplied by the vendor industry (individual recommendations on the size and design of the various components), the envelopes for each type and size module were established.

Although it had originally been planned to have three distinct flow sizes for the majority of the components, it has always been apparent that if the smallest feasible envelope can be designed to operate satisfactorily over the entire 0 to 25 GPM range, the resulting fabrication cost savings and simplified logistic problems of having only one component size would be definitely desirable. As a result of new and better design approaches in competitive bidding by the vendors participating in the modular program, it has been possible to utilize certain unit envelopes whose advantage of carrying more than one flow range offsets the small dimensional reduction necessary for separate flow sizes.

The following is a list of components which were designed, fabricated and tested; the number of flow sizes developed; the number of parts procured, and the number of vendor contracts required for each component.

Component	Flow Sizes	Total Units Req'd Each Component	Number Vendor Contracts Req'd
Check Valve	3	(9)	3
1-way Restrictor	1	(3)	1
2-way Restrictor	1	(3)	1
Shuttle Valve	3	(9)	3
Thermal Relief	3	(9)	3
Pressure Switch	1	(3)	1
Priority Valve	3	(9)	3
Relief Valve	2	(6)	2
2-way Solenoid Selector	1	(6)	2
3-way Solenoid Selector	2	(9)	3
4-way Solenoid Selector	3	(9)	3
Solenoid Sequence Valve	2	(6)	2
Filters	3	(9)	3
Total Sizes	28	Total Units 90	Total Vendor Contracts 30

Specifications patterned in MIL specification form have been prepared for each of the above components and are based on an assimilation of data from all sources as to the latest requirements for a 4,000 psi, 450°F component and include a complete vibration test under functioning conditions, induced fluid contamination, and endurance cycling over a time-temperature spectrum.

The idea of a cartridge component is by no means unique but what this program has provided is sufficient time and effort to be applied to the thinking out of the general approach and consideration of all design factors for standardizing cartridge components.

The inclusion of the hydraulic industry in this venture has enabled the program to produce actual, useable, fully qualified Type III system components, but has also produced a less tangible although just as beneficial result; and that is the implanting of the principles of modular design in a good portion of the industry along with the capability of producing modular components for all type system requirements.

The complete report of module component development is given in Part II of the final report of the modular hydraulics program.

2. Package Development

This phase of the program was concerned with developing reliable methods for packaging modular valves in single manifolds for use in a 4,000 psi, 450°F hydraulic system. A primary product of this phase was a specification in MIL specification form giving recommended practices for packaging hydraulic sub-systems and designing the necessary manifolds. The contents of the specification were based on design and testing of actual hardware.

The general approach taken in solving the packaging problems was as follows:

a. Airplane and missile hydraulic sub-systems were studied for package feasibility. In order to get a "feel" for packaging and the advantages that result, several typical hydraulic sub-systems were packaged.

b. For the purpose of testing modular components and manifolds under environmental conditions, hydraulic sub-systems were designed to utilize each type of modular component with as many as 5 units being packaged in a single manifold. Because of the limited number of components which were available, only 4 packages were designed.

c. Techniques of servicing packages were studied. Is it best to remove individual valves or the entire package for servicing? How do you remove modular valves without losing system oil? How do you remove packages without losing system oil? These and other problems were studied and recommendations made.

d. Methods that make manifold design easier and lighter, and less expensive, were studied and recommendations made. This included items such as the use of permanent plugs, castings, and forgings.

e. Although modular valves were qualified by the vendors, testing was also performed on the packages as a unit.

The four hydraulic test packages were designed and fabricated specifically in order to:

a. Gain experience and insight into the design requirements and fabrication problems inherent in packaging several modular components together in one common housing. Designs were made for an optimum "weight and space" requirement in order to demonstrate the compactness resulting from the modular concept and to determine the materials and material shaping processes that were required to achieve this result.

b. Gain experience in the characteristics of a 450°F, 4,000 psi system when these packages were individually and concurrently tested in a high and low temperature test system.

Packages fabricated were as follows:

a. Titanium Manifold - Housing for 3-way selector valve, 2-way restrictor, and a check valve.

b. Actuator with Modular End Cap - End cap houses a pressure relief valve, a shuttle valve, and a 1-way restrictor.

c. Stainless Steel Manifold - Housing for a priority valve, thermal relief valve, pressure-operated shut-off, solenoid-operated sequence and a cartridge 4-way-3-position selector valve.

d. Stainless Steel Manifold - This package is in two parts: one section is intended to be mounted permanently to the structure and contains the plumbing connections; the other section contains the valves, is removable and mounts to the stationary section. Both parts contain internal shut-off features. The removable section houses two filters, a solenoid-operated shut-off valve, pressure relief valve, and a pressure switch.

The packages resemble castings or forgings in order to obtain more realistic design and test results and to present a truer likeness of production-type packages than could be gained by "block type" machined manifolds.

The use of a hydraulic package is certainly not new. There have been many package designs in aircraft and missiles. Also, the commercial hydraulic industry has used this technique to advantage for some time. This program did not attempt to standardize package designs but does provide established components and a workable philosophy for use in packaged designs. However, when any new system is approached with the idea of using the package technique exclusively, there will be limitations that must be considered.

a. More design time and effort will be required from concept to production. This factor will be doubly limiting if pre-established cartridge components are not available.

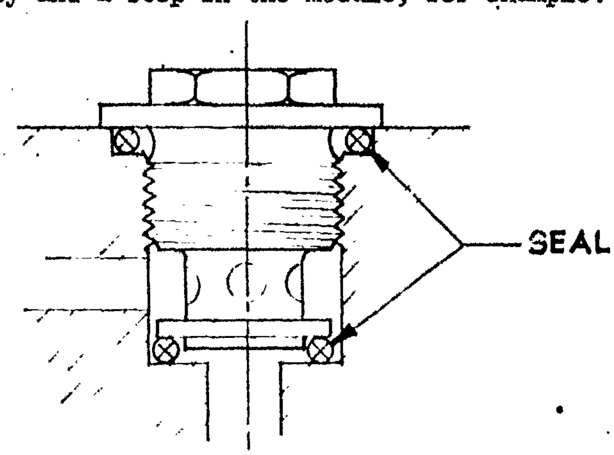
b. Produces a rather inflexible design which cannot be changed readily. The gamble of waiting until the actual system is operating to solve minor problems and prove system integrity will not be tempting. System tests will have to be simulated and proved long before fabrication.

c. Initial cost will be high. This includes the lead time required and is only true because the savings that a highly reliable and easily serviced system affords is difficult to estimate.

The complete report of package design and development is given in Section III of this volume (Part I).

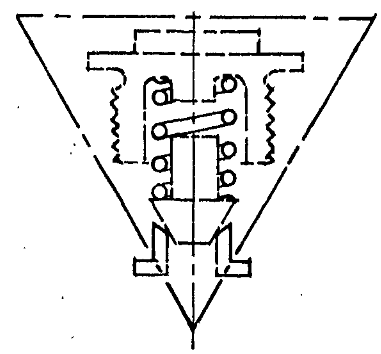
3. <u>Metallic Seal Program</u>

All of the modular components with one exception will utilize a face type static seal that will fit between a step in the manifold cavity and a step in the module; for example:



This type of seal is based on the principle of controlled compression and the use of the step design allows the compression force to be exerted by the threads of the module and the manifold on installation of the unit. This simplifies assembly and allows a one-step installation and removal procedure.

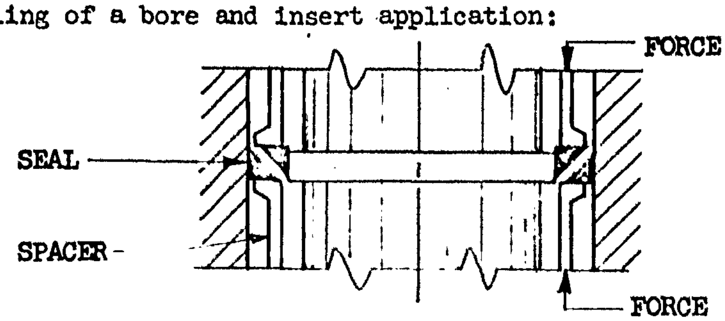
The use of the step design in no way compromises the design of the majority of the units, because when the component designs are tailored for a cartridge application their functional shape is triangular in form.



The exception to this pattern is the 4-way selector valve. When converted for a cartridge application, this unit, due to the rather elongated sleeve and slider for opening and closing several passages, is basically rectangular in form. To adapt a step seal design to this valve would be compromising the design and would result in considerably more mass than required due to the stair-step shape. Therefore, a metallic seal that permits sealing between the rod-shaped O.D. of the valve and the bore of the cavity is required.

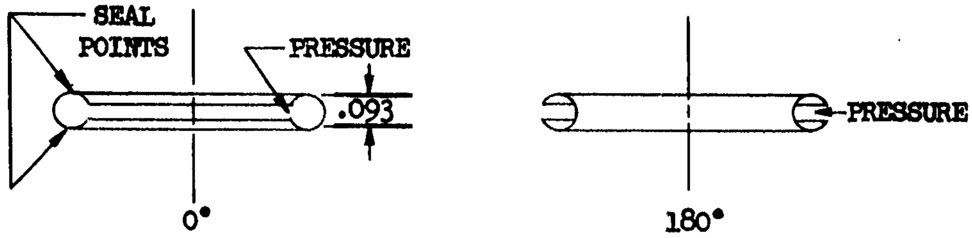
Subsequently, there were two configurations of metallic static seals developed and qualified:

a. Static Wedge Seal - This is a two-part, solid-ring metallic seal that is mechanically stressed for static, two-way sealing of a bore and insert application:



This seal is used in the modular cartridge 4-way solenoid selector valve and requires 6 pairs of wedge seals.

b. Hi-Ceal (C-seal) - This is a one-piece, formed, stainless steel ring with a C-shaped, 0.093 inch cross-section. Configurations utilized for the modular program were :



Two-way sealing is accomplished by combining the two configurations.

Primarily, the decision to use metallic static seals in this program and Type III systems was based on the temperature requirements; however, there are some additional advantages of the static metallic seals being developed.

a. The face-type seal is fabricated by a forming process and would be relatively inexpensive in production quantities.

b. A metallic seal can be used with a variety of fluids without the danger of material decomposition.

c. The seal, being metallic, is less subject to damage in handling and installation.

d. Tests performed have indicated that the probability of ever having a catastrophic failure is remote.

The complete report of metallic seal development is given in Section II of this volume (Part I) of the modular program final report.

4. Integrated System Design

This phase of the program was concerned with the means of transmission of fluid power from the source to the modular packages and to the actuation devices, and has as its end objective the creation of fluid passages so reliable that no maintenance is needed for the life of the vehicle. The work area encompassed all sizes and types of structural members including items such as structural bolts, rotating joints, and linkages.

The work in this phase was basically divided into two sections:

Engineering Studies - This area was devoted to a continuing study and research into the possible methods of carrying fluid in structural members, processes and development which will be required for the various members, processes and development which will be required for the various methods, and design layouts of the more promising methods for realistic applications.

Hardware Development - This area was concerned with designing, fabricating and testing various configurations of fluid-carrying structure. Specific items which were tested for feasibility included rotary joints, tube joints, connectors for joining fluid-carrying members, package mounting pads, and a fluid-carrying ball-joint for actuator use.

The integrated system concept is much broader in scope than just the creation of a few random pieces of fluid-carrying structure or the selected use of a few modular packages. In order to realize fully the advantages such an approach offers, it will be necessary to consider fluid power transmission lines and component packages during the preliminary design phase and integrate design of these items with the structure so that both structure and systems are complete together.

The complete report of studies and hardware development for the integrated system design concept is given in Part III of the final report of the modular program.

G. General Discussion

1. Development Summation

If the problems encountered in the various development areas of the program were categorized, it would be apparent that the majority of the difficulties experienced in this initial research into modular hydraulic systems were due to the temperature and pressure requirements of the program, and would have existed regardless of the philosophy of design. The basic advantages of the modular approach, particularly in reliability improvement, have remained valid throughout the program. However, the materials, processes, and techniques required to produce a 4,000 psi, 450°F system need continued development.

This program is by no means the complete solution to the approaches offered here. The program was intended to be the first step in modular or integrated system design concepts and to show that there are ways of improving system design and reliability by considering all factors of the system; function, components, installation, and connection. An improved design plan and proofs of feasibility are what this program offers. The final solutions, proofs and benefits will be in actual application use.

2. Direction and Coordination

The development effort of the modular program was directed by the Applied Research and Development Section of the Aeronautics Division. The program was conducted as a major project and utilized the capabilities of all major engineering sections of the division.

The program was coordinated with the fluid power industry by regular presentations to the SAE Committee A-6 for Aerospace Hydraulic and Pneumatic Components and Systems. Detail progress reports were made on a quarterly basis to the Airborne Equipment Division of the Bureau of Naval Weapons, Aeronautical Systems Division, Wright Field, and to the High Temperature Panel of Committee A-6 which acted as an advisory board for the program. The eight quarterly progress reports issued were numbered as follows:

Quarterly Report	Vought Aeronautics Number
1st	E9R-12062
2nd	E9R-12075
3rd	AER-E9R-12449
4th	AER-E9R-12467
5th	AER-EOR-12636
6th	AER-EOR-12909
7th	AER-EOR-13104
8th	AER-EOR-13270

II. METALLIC SEAL DEVELOPMENT

A. INTRODUCTION

At the time the modular hydraulics proposal was submitted to BuAer, an upper temperature limit of 400°F was planned. On January 19 and 20, 1959, a conference was held at Chance Vought for the purpose of coordinating the development work being done under the modular hydraulic program with the weapons systems being designed by Air Force and Navy contractors. One result of this conference was the decision to establish an upper temperature limit of 450°F for the modular hydraulic program in order to make it compatible with weapons systems under development.

With the upgrading of the maximum temperature to 450°F, it was felt that elastomeric seals presently available were marginal and that only metallic seals should be used. A brief, preliminary industry survey indicated that the state-of-the-art in metallic seal development was not sufficiently advanced to provide the one best static seal for all modular applications. Several designs of metallic seals for boss and face seal type of applications existed. However, the capabilities of these designs had not been established. Moreover, certain design features of existing seals, such as close tolerances, fine surface finishes and high seal installation forces (torque), appeared undesirable.

Because of the importance of the seal to the success of the modular hydraulics program, a metallic seal development program of considerable magnitude was undertaken. The detail objectives of the program were to:

1. Establish a specific metallic seal design or designs that would be adequate for the successful completion of the modular hydraulics program.

2. Qualify the metallic seal for service usage.

3. Develop a vendor source of metallic seal supply.

To carry out these objectives, it was felt that promising seal designs must be procured, tested and evaluated by Chance Vought. The most promising designs would be selected for development tests by the seal manufacturer. Chance Vought would work with the vendor during development to expedite the program and to investigate all requirements and problem areas. Once all the "bugs" are worked out, the vendor shall be required to qualify the metallic seal to the modular program requirements to establish reliability for service usage.

B. METALLIC SEAL DEVELOPMENT

1. Desirable Qualities of the Metallic Seals

Before starting detail evaluation of metallic seals, a listing of problem areas and seal requirements that affect seal development was prepared. Later in the seal development program it will be noted that several compromises of ideal requirements had to be made, which resulted in a suitable rather than an ideal metallic seal eventually being qualified. The optimum seal requirements are as follows:

Proper Size - Since one objective of the modular program is to miniaturize hydraulic components, it is necessary that the seal be small. The establishment of a cavity in which the metallic seal must operate will be a compromise between what the seals require and what the hydraulic components require.

Sealing Ability - The seal must seal hydraulic fluid from -65°F to +450°F throughout a pressure range of zero to 6,000 psi. This if, of course, the prime purpose of the seal, and if it cannot perform this function all other seal requirements become unimportant.

Compressive Force - Force to compress the seal during installation must not be excessive. Excessive forces result in excessive modular component installation torques, which in turn can result in thread failures. It will later be seen that this requirement caused considerable investigation in an effort to reduce compressive force while maintaining sealing capabilities. Items affecting compressive force are configuration, seal material, tensile strength (hardness) and the seal wall thickness.

Seal Removal - The seal must be easy to install and remove from its cavity. This can be controlled by design considerations of both the cavity and the seal.

Reverse Pressure Sealing - The design of modular components requires that the one-way seals withstand up to 1,000 psi in the reverse direction.

Squeeze - Preliminary studies of seal cavity machining tolerances indicate that the initial squeeze which a seal may be subjected to in any one cavity may vary .008 inch. An additional squeeze variation will be the result of seal tolerances. This requirement is very stringent on the structural properties of the seal. It also increases the maximum squeeze force of some installations by requiring greater seal deflections.

Diametral Back-up - This back-up should not be too close since the seal may expand slightly and bind in the cavity. Yet close back-up will likely be required to provide adequate seal life under

impulse testing. Machining tolerances can be held to .001 to .003 inch on the back-up dimension in the cavity. This small amount of variation from one cavity to the next is not likely to cause problems.

Re-Useability - Re-useability will depend on the amount of springback the seal exhibits after compression and on the variation of squeezes resulting from machining tolerances. The implication is that the seal would have to spring back 100% for unrestricted re-useability. If this is not feasible, there should be enough springback to tolerate a reduction in squeeze of about .001 inch caused by spreading the cavity sealing under high pressures in face seal applications.

Cost - The method of manufacturing the metallic seal should suggest that the cost of the seals would not be prohibitive under mass usage.

Wear - Neither the seal nor the cavity sealing surfaces should exhibit excessive wear under impulse testing. It will later be noted that wear does occur and has not been entirely eliminated. This presents a problem both from a sealing and a contamination standpoint.

Installation - The seal and seal seat must withstand the "scrubbing" and "twisting" action that takes place when the modular component is torqued down on the seal.

Endurance - When installed under adverse conditions, the seal shall be capable of withstanding 200,000 pressure impulse cycles of 0-6,000 psi at a temperature of 450°F.

2. Procurement of Metallic Seals for Preliminary Testing

Early studies concerning packaging of hydraulic components indicated the shape of the components and correspondingly the type of seals required. The type of seals required for packaging and component sealing were determined to be boss, face and radial. It was found that many metallic face seals were already on the market and that some work had been done with metallic radial seals.

Interest was stimulated in the development of metallic seals for the modular hydraulics program by releasing a general procurement specification, CVC-M.H.2, to approximately 30 seal manufacturing companies throughout the United States. (This specification is included in the appendix.) The specification was a result of early studies and outlined tentative information on design, manufacturing, application and testing considerations relative to the types of seals required. Applications of the boss, radial and wedge are shown, but no dimensions are given. The endurance test which requires the seal to withstand 200,000 pressure impulse cycles remained as described throughout the program. The impulse requirement was taken from hydraulic

fitting requirements, with the idea being that the seals should be at least as rugged as the fittings. Although the requirements were rather stringent, the response to specification CVC-M.H.2 was excellent. Of thirty manufacturers contacted, twenty responded. (See Appendix II-2) Twelve different types of seals appeared applicable to the modular hydraulic program and were procured for evaluation.

3. Preliminary Test Preparations

The twelve types of metallic seals procured for evaluation required that test cavities for each type seal be designed and fabricated by Chance Vought. The test manifolds shown on drawings TL3618 and TL3628 are included in Appendix II-3. Other test manifolds were manufactured for testing the metallic boss seals. This fixture is also included in Appendix II-3.

A test set-up was designed and assembled at the high temperature facilities to give the pressure impulse and temperature requirements specified in CVC M.H.2. A schematic of this set-up and how it works is given in Appendix II-3. Also shown is the glass standpipe arrangement used to measure small amounts of leakage at high temperatures. By trapping the leakage in these standpipes, no vaporization of the fluid takes place. The glass standpipes also afford ready visual measurement of the leakage taking place during testing.

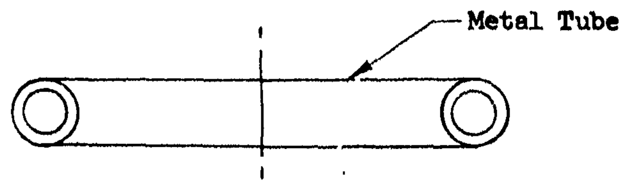
A "General Test Procedure for Metallic Seals" was prepared using specification CVC M.H.2 as a reference. This procedure is included as Appendix II-5 and should be referred to for the complete procedure used to test both boss and face seals.

4. Preliminary Testing of Seal Designs

A summary of the test results for each type of metallic seal tested is given below. For detail results refer to Appendix II-4. It will be noted that final selection was not "cut and dried" because several of the seals performed very well. All seals were tested per the general test requirements in specification CVC M.H.2 (see Appendix II-1). Each seal was evaluated using the test requirements described earlier as a guide (i.e., proper size, sealing ability, compressive force, reverse sealing, etc.). The seals tested are listed in alphabetical order for convenience. It must be stressed that these various seals were being tested to very particular requirements and failure to meet these requirements does not mean that these seals would not perform satisfactorily in other applications.

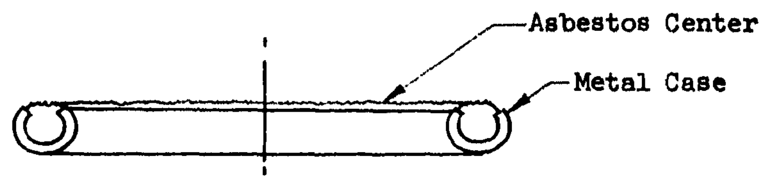
Advance Seal - The Advance Seal is essentially a hollow metal O-ring. Five sizes ranging from 0.375 to 1.750 inches in diameter were tested. Of the sixteen tested, eight were teflon-coated and eight were silver-plated. The teflon-coated seals did not perform satisfactorily. The silver-plated seals performed well during static leakage tests, pressure

impulse tests, and reverse pressure sealing. These preliminary tests indicated that it may be adaptable to the modular program.



ADVANCE SEAL CROSS-SECTION

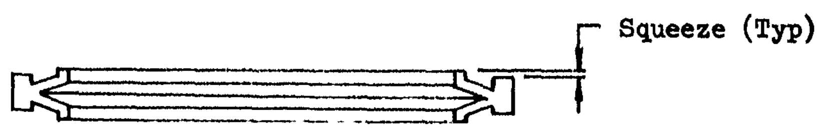
Aero Gasket Seal - The Aero Gasket Seal consists of a thin metal case enclosing an asbestos fiber center. The seals performed



AERO GASKET SEAL CROSS-SECTION

well under static pressures, but leaked during the pressure impulse tests. Spring-back tests showed this seal to have very little resiliency. It was eliminated from further consideration.

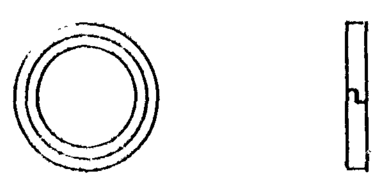
Cadillac Gage Seal - This seal is machined from stainless steel. Three of the four seals tested performed satisfactorily. The



CADILLAC GAGE SEAL CROSS-SECTION

seals withstand reverse pressure very well. However, because it was determined that the modular seal cavity machining tolerances will cause squeeze to vary as much as 0.010 inch, this seal was eliminated from further consideration. It was felt that the seal could not tolerate squeezes that varied between 0.006 to 0.016 inch.

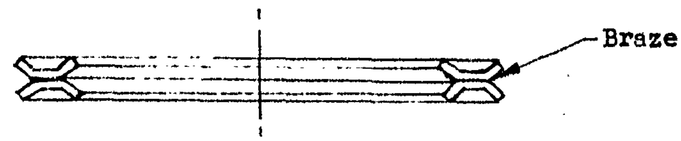
Double Seal - This seal is a metallic seal proposed for radial application. The seal consists of two parts, an inner steel expander



DOUBLE SEAL

and an outer steel ring, both having rectangular cross-section. The seal leakage was excessive for a static seal.

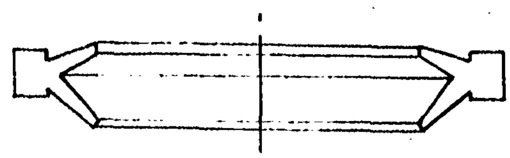
Chance Vought X-Seal - The X-Seal is composed of two formed pieces brazed together to form a ring having an X shaped cross-section. The reason for this design was to establish a relatively low



X-SEAL CROSS-SECTION

cost, effective seal that could be used for both two-way and one-way applications. Problems occurred in fabricating the seal, and its sealing capabilities were not particularly outstanding. Tests showed that compressive forces were too high for program use.

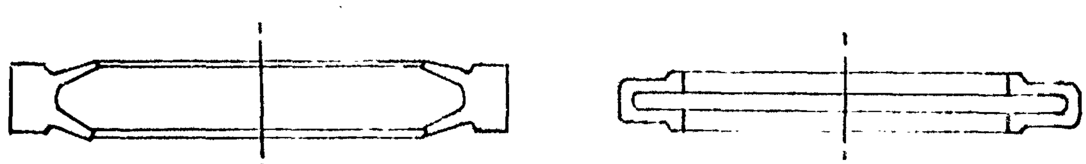
Harrison Boss Seal - This seal was designed to seal in an AND 10050 port. Because a boss seal is a very necessary part of this program, considerable work was done with the Harrison Boss Seals to determine



HARRISON BOSS SEAL CROSS-SECTION

their effectiveness and to gain experience in using this type of seal. Detail evaluation is included in Appendix II-4 . In general, it was discovered that: (a) high torque requirements cause rotation of the fitting; (b) goldplate outperformed teflon; and (c) although not as reliable as desired, it could be used.

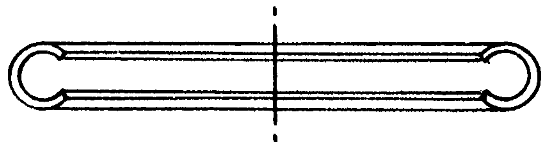
Harrison Face Seal - Two different configurations of the face seal were tested. Of the sixteen tested, eight were teflon-coated



12130 Face Seal CROSS-SECTION 12110 Face Seal

and eight were gold-plated. The teflon coat peeled away and was considered of no practical value when subjected to the 450°F environment. The gold plate held up better than the teflon, but gold-plated seals as well as teflon-coated seals leaked. Their over-all leakage performance eliminated them from further consideration.

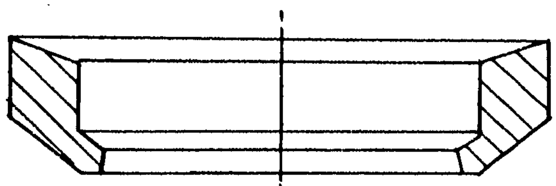
Hi-Ceal Face Seal - Preliminary testing on the Hi-Ceal was very encouraging. Two out of the first three tested passed the 50,000 pic[1] test with no failure. Reverse pressure sealing was good. Five other seals



HI-CEAL CROSS-SECTION

were subjected to static pressures up to 6,000 psi. There was zero leakage. The force to compress the Hi-Ceals during installation was large. However, from the design of the seal, it appeared the force could be reduced to an acceptable value. Another problem which was felt to affect test results was the lack of uniformity of the finished seals. In general, it was felt that this seal showed much promise in meeting sealing, cavity, installation torque, reverse sealing and endurance requirements of the modular program.

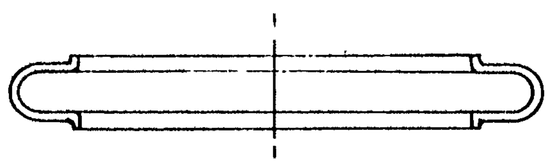
Navan Boss Seal - Two each of -6, -10, -16, -20 and -28 size seals were tested. The -6, -10 and -16 were subjected to 61,000 pic



NAVAN BOSS SEAL CROSS-SECTION

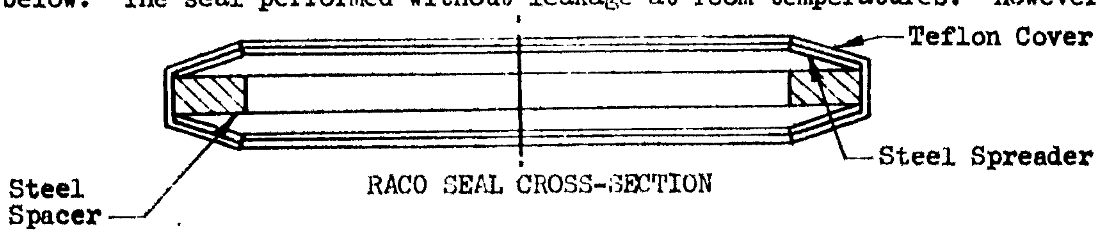
without evidence of leakage. The -20 and -28 were leaking after 11,800 pic. The torque to prevent leakage on these seals was considered must too high for practical useage.

Omega Seal - Only one of these seals was tested. The seal performed very well, but further investigation was not pursued because additional test samples could not be obtained.



OMEGA FACE SEAL CROSS-SECTION

Raco Seal - This face seal has a teflon cover as shown below. The seal performed without leakage at room temperatures. However,

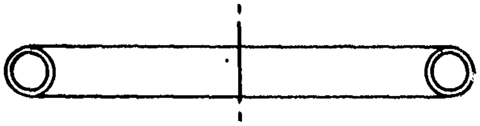


RACO SEAL CROSS-SECTION

[1] pic - "pressure impulse cycles"

as the seal was being heated during pressure impulse cycling, the teflon extruded at 380°F. No further tests were performed.

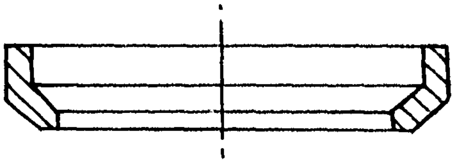
Toruseal - Five of the eight Toruseals tested had zero leakage during normal pressure impulse testing. The seal also performed



TORUSEAL CROSS-SECTION

acceptably during reverse pressure tests; however, the torque to compress the seals was considered too high for the modular program application.

WADC Boss Seal - A torque of 3,280 in./lbs. was required to seal one -16 seal when it started leaking after 27,820 pic. The two



WADC BOSS SEAL CROSS-SECTION

-20's and the other -16 required 1,860 in./lbs. to seal when they began leaking after 47,217 pic. Although sealing performance was good, two disadvantages dismissed further testing. One is that torque is considered high; the other is that a pre-setting tool is required. These seals have performed very well, however, in the modular program impulse and functional test stands where it was possible to pre-set and torque the seals properly.

5. Selection of Hi-Ceal for Development

While preliminary testing was being accomplished on the various metallic face seals, work was also being done on the problem of defining the seal cavity. As the result of coordinating seal requirements with modular component and cavity dimension requirements, tentative seal cavity charts were determined. The charts show cavity requirements for one-way and two-way seals and include cavity dimensions for 3/16 inch I.D. to 3.00 inch I.D. seals. The charts in their final form are shown in specification CVA 2464 in Appendix II-5 . Although the charts were determined early in the program and later coordinated with North American, very few changes were necessary.

Several seal designs were eliminated from further consideration by the firm definition of seal cavity requirements. Some companies could not reduce their seal size to meet modular component requirements, and others did not feel that they could assume the expense of a re-design. Of the seals that were tested and that met cavity requirements, the Hi-Ceal showed the most promise of meeting all of the seal requirements

of specification CVC M.H.2. With additional information learned from preliminary testing, a firm procurement specification, CVC 2464 (Appendix II-5), was prepared. With Phase I of CVC 2464 spelling out the requirements, a contract was awarded to High Pressure Engineers, Inc., for a development program to adapt the Hi-Ceal design to both face seal and radial seal modular applications. The radial seal effort was later dropped when it was determined that installation forces were too high.

Although a feasibility contract was awarded to High Pressure Engineers, Inc., for continued development of their seal for modular component sealing, Chance Vought continued to investigate the possibilities of finding a better metallic face seal. In addition, Chance Vought undertook much of the Hi-Ceal development test work to expedite the seal program so that seals would be ready for use in the modular hydraulic components that were being developed.

Test work on metallic seals eventually boiled down to a concentrated effort by High Pressure Engineers, Inc., and Chance Vought to get the Hi-Ceal developed and qualified. It is desirable to have more than one metallic static seal developed and qualified for modular hydraulic use, but funds to "sponsor" more than one program were not available.

6. Development of the Hi-Ceal

Several of the problems associated with developing the Hi-Ceal were handled in a straightforward manner. Other problems required considerable testing before final conclusions were made. Problems such as seal and seat wear, seal fatigue failures, and reducing seal compressive forces to an acceptable value were particularly troublesome. Satisfactory solutions were obtained for all problems, however wear and excessive compressive force were two problems where further improvement would be desirable. The problems were treated individually and are noted below.

Sealing Ability - The seal must be able to seal under the conditions set forth in specification CVC 2464. During the investigation of all problems, sealing ability was monitored and was the basic requirement for successful results.

Size - Because one of the efforts of the modular hydraulic program was to miniaturize hydraulic components, it was necessary that the seal be small to be compatible with the miniaturization effort. As information on component design became available, the shape and size of the components were optimized. From preliminary tests performed on metallic seals, it was learned what seal cavity size would be acceptable to several different seal designs. The above information was used to establish a standard cavity size in which a metallic seal used with modular components would have to operate. The seal cavities established are shown in specification CVC 2464 in the appendix. It was expedient that size be established early in the program so that design and development of the modular components would not be hindered. This was accomplished satisfactorily. The preliminary tests, early establishment of cavity size, and experience that High

Pressure Engineers, Inc., had with the Hi-Ceal, helped to establish the 0.015 wall thickness, 0.093 thickness, 304 material Hi-Ceal for interim use in component development.

North American Aviation became interested in the Hi-Ceal early in the program and expressed a desire to coordinate development of this seal with Chance Vought. The seal was being developed in sizes up to 3 inches under the Chance Vought program. North American planned to continue development up to 8 inches. The seal cavities in CVC 2464 were coordinated with North American and finalized with very little change.

This work resulted in reducing the cross-section thickness of the Hi-Ceals from 0.125 inch to 0.093 inch. Later in the program it was

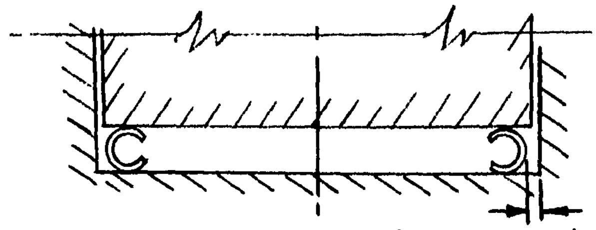
HI-CEAL CROSS SECTION

shown by calculations that reducing the thickness was one of the factors which limited further reduction of seal compression force. These calculations are shown in High Pressure Engineers, Inc., final Qualification Report.

<u>Reverse Pressure Sealing</u> - Under terms of the development contract, High Pressure Engineers was to develop the Hi-Ceal for both one-way and two-way modular component applications. In modular component applications where a one-way seal is required, the seal must also resist some back pressure, or reverse pressure. Early tests on 1.00 x .093, 304 material, 0.015 wall thickness withstood 5,950 psi reverse pressure before the seal collapsed. Another Hi-Ceal withstood 45,000 reverse pic at 450°F, but leaked during latter stages of the test.

The conclusions that were reached from the relatively few reverse pressure tests were that the one-way Hi-Ceal is good for reverse sealing only up to the required 1,000 psi. It did not perform well enough to replace the two-way Hi-Ceal.

<u>Seal Removal</u> - Tests were run varying the back-up groove diameter, squeeze and seal wall thickness to determine the radial clearance required for easy removal of the seal after maximum squeeze and 6,000 psi proof pressure. The initial tests indicated that a radial clearance, prior to installation and pressure, of approximately 0.007 inch is required for the Hi-Ceal to drop freely out of the test manifold. Based on these tests, radial clearance was established as 0.005 to 0.007 inch. Although other development problems later in the program required the evaluation of closer back-up, 0.005 to 0.007 inch was the value used by High Pressure Engineers in the Hi-Ceal qualification tests.



Modular component vendors used the 0.015 wall, 304 material Hi-Ceals throughout their development programs with a radial clearance of 0.005 to 0.007 inch. Only three cases of the seal sticking in its cavity were reported.

Of the development tests in which radial clearances as low as 0.0005 inch were used, in an effort to reduce wear, very few cases of the seal sticking occurred. Two cases were recorded in which a force of approximately 50 pounds was required to remove the Hi-Ceal.

Diametral Clearance - As previously discussed, one of the parameters which control the clearance between the Hi-Ceal and the cavity is that the seal should be easily removable from the cavity after being subjected to maximum squeeze and proof pressure. To satisfy this requirement, early tests established 0.010 to 0.014 diametral clearance as being satisfactory.

Later in the program tests at Chance Vought indicated that one of the ways to increase the seal life during impulse testing was to use closer back-up clearance. In optimizing back-up clearance for the 0.015 wall, 304 material Hi-Ceal, it will be noted from groups 2 and 3 in Table II-1 that good performance is obtained with close back-up clearance whether squeeze is minimum or maximum. With back-up clearance increased to around 0.012 inch, good performance is obtained when the squeeze is toward maximum, but performance suffers when squeeze is minimum and back-up is large. It was concluded for the tests at Chance Vought, performed on the 0.015 wall, 304 Hi-Ceals, that the specified diametral back-up clearance should be in the range of 0.003 to 0.006 inch. This allows good impulse performance without causing Hi-Ceals to bind in their cavities.

Tests performed at High Pressure Engineers, Inc., did not indicate any significant improvement of performance by reducing diametral clearance. Since satisfactory results were also being reported by vendors using 0.010 to 0.014 inch diametral clearance in value qualification tests, it was decided to qualify the Hi-Ceal with the original diametral back-up clearance. Hi-Ceals successfully passed qualification using 0.010 to 0.014 inch diametral back-up.

Squeeze - A study of the machining tolerances, which is included in the appendix, shows that the initial squeeze which a seal may be subjected to in any one cavity may vary 0.012 inch. An attempt was made to establish the squeeze range between 0.004 and 0.016 inch. Keeping the required squeeze low had two advantages: it reduced installation torque and it allowed greater seal springback. However, tests using 0.004 inch squeeze did a poor job of sealing during both room and 450°F static pressure tests. The minimum squeeze was raised to 0.006 inch to increase the mechanical force between the seal and the seats. Considerable improvement was noted. The maximum squeeze was accordingly increased to 0.018 inch. Greater squeezes were not considered because of the effect on installation torque and possible structural failure from excessive deflection. The deflection versus compression force curves in the appendix show how compression force increases with greater seal squeeze. These curves also point out that when deflection exceeds about 0.006 inch, further increase in deflection does not appreciably affect springback.

Development and qualification tests verified that the squeeze range of 0.006 to 0.018 inch for the Hi-Ceal is seal is satisfactory. This Hi-Ceal squeeze range has also been used by vendors during development and qualification testing with satisfactory results.

Re-useability and Springback - Because of the large variation of squeeze (0.006 to 0.018 inch) that a seal can be subjected to in different cavities, it was considered unlikely that the Hi-Ceal could be developed to the extent that it would have unrestricted re-useability. Unrestricted re-useage for a metallic seal used with modular components implies 100% recovery or springback after being squeezed and pressurized. For example, consider a Hi-Ceal that was installed in a cavity that subjected the seal to an 0.018 inch squeeze. To put this seal in a cavity where squeeze was only 0.006 inch would require the seal to spring back at least 0.012 inch to make contact with the cavity seats. Another 0.006 inch would be required in order to meet the 0.006 inch minimum squeeze requirement. Conversely, putting a seal that had been squeezed 0.006 inch into a cavity that normally squeezes a Hi-Ceal 0.012 to 0.018 inch is entirely satisfactory. In service applications of the Hi-Ceals, it is not considered practical to reuse them. However, for laboratory useage, the cost of the seals may make it worthwhile to consider re-useage.

Chance Vought, High Pressure Engineers, and most of the component vendors, have had good success in reusing Hi-Ceals in the same cavity; not 100% success, but enough to make it a financially worthwhile consideration.

Early tests with the Hi-Ceal established that the Hi-Ceal took a permanent set when squeezed to the required deflections of 0.006 to 0.018 inch. Force deflection curves for various Hi-Ceal materials, both as formed and annealed, are shown in Appendix II-7 . Figure II-1 represents typical results. These curves show that the seal material yields when deflection increases beyond 0.004 to 0.005 inch. Because of this, little springback can be expected. The following curves show a springback of about 0.002 inch whether the squeeze is 0.020 or 0.010 inch.

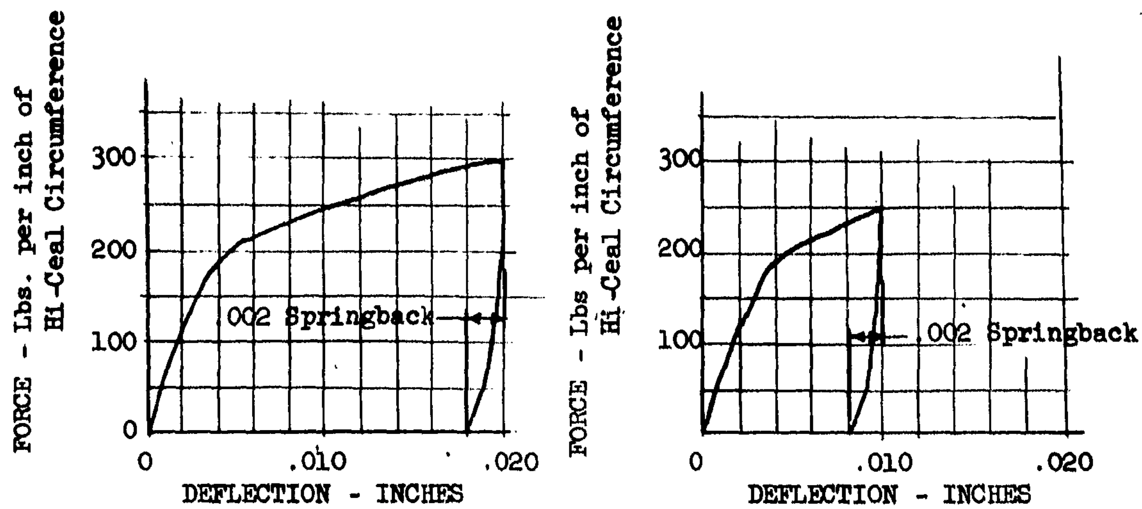


FIGURE II-1

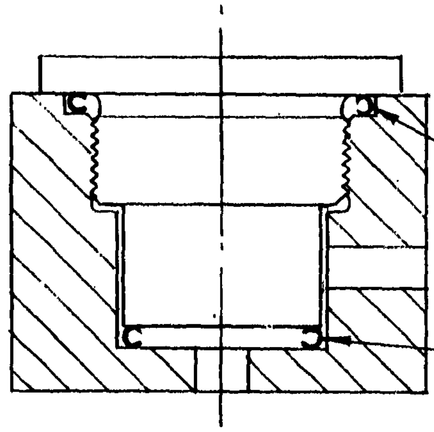
TYPICAL FORCE DEFLECTION CURVES FOR 0.015 WALL, 304 HI-CEAL

From curves in the appendix it can be seen that other Hi-Ceal materials with different wall thickness showed greater recovery properties than the 0.015 inch wall, 304 material Hi-Ceals. However, these materials did not seal as well as the 304 material. Both AM350 and Inconel X Hi-Ceals had a springback greater than 0.004 inch. In no case is springback great enough to allow unrestricted re-useability.

Springback has its greatest value in modular sealing applications by making the seal tolerant of squeeze changes that may occur during pressurization or temperature changes. In addition, springback allows some reuse of Hi-Ceals in their same cavity.

Installation Torque - One of the basic requirements in selecting a metallic seal was that the force to compress the seal must not cause excessive modular component installation torque.

During the preliminary testing and early development stages of the Hi-Ceal, Chance Vought procured several 0.019 wall, 304 material Hi-Ceals to further investigate the possibility of using zero degree Hi-Ceals for two-way sealing. A 2.750 inch diameter seal was being used as the secondary seal, and a 1.00 diameter seal was to be reverse pressure tested. As the plug was tightened, it became apparent that a very large torque was required to compress the Hi-Ceals to the required deflection. A force versus deflection test was performed on the unused Hi-Ceals and it was learned that the compression force was two or three times greater than earlier tests by High Pressure Engineers had indicated. The discrepancy was attributed to two things. First, the earlier force versus deflection tests by High Pressure Engineers had been performed with 0.125 inch cross-section Hi-Ceals, whereas 0.093 inch cross-section seals were now being used. Secondly, the 0.093 seals being tested were formed from Hi-Ceals that were originally



TEST BLOCK

0.125 cross-section. Because 304 is an austenitic steel, it can be work-hardened during the forming process used by High Pressure Engineers in manufacturing the Hi-Ceal. It was suspected that the additional forming caused the strength of the Hi-Ceals to increase, thus increasing seal compressive force.

Several courses of action were pursued in an attempt to reduce modular component installation torque: (1) Some consideration was given to returning to the 0.125 cross-section Hi-Ceal. (2) Annealing the Hi-Ceal material both before and after forming was tried. (3) Using a thinner wall Hi-Ceal was investigated. (4) Different Hi-Ceal materials were tested. (5) Plating or coating the Hi-Ceal was tried in an attempt to reduce friction between Hi-Ceal and cavity seat. (6) Coating the threads of the modular components with thread lubricants was investigated as another means of reducing friction. Approaches (2) and (3) precluded further tests using zero degree Hi-Ceals for full two-way sealing.

While the aspects of returning to the 0.125 cross-section Hi-Ceal were being studied, High Pressure Engineers reported that they had run tests on thinner wall Hi-Ceals in the annealed and as-formed conditions, and that these Hi-Ceals had withstood 10,000 psi static pressure and had been impulsed 200,000 times at room temperature. Since these reports sounded encouraging, and to return to the 0.125 cross-section would cause some component design changes, it was decided to maintain the already established 0.093 cross-section.

Calculations were made which indicated that the force to compress the Hi-Ceal ideally should not exceed approximately 100 pounds per inch of seal circumference. This figure was arrived at by estimating what would be an acceptable component installation torque value, then solving for what seal compressive force gave this value. The acceptable installation torque was based on values that could be handled conveniently. These calculations, which are included in Appendix II-8 , also show that the majority of the torque applied to a component is lost in the thread friction and in friction between the Hi-Ceal and cavity seats. These calculations

suggest two approaches to consider in reducing torque; namely, reducing thread friction and Hi-Ceal surface friction. These two approaches are mentioned in an above paragraph and will be discussed later.

Compression tests were performed on 1.000 diameter, 0.093 cross-section, 0.015 and 0.019 wall Hi-Ceals which had been annealed. A detail discussion of the aspects of annealing the Hi-Ceals is included in the "Materials Report," Part IV. The test results showed that both the 0.015 and 0.019 annealed Hi-Ceals still had excessive compressive force. The 0.015 wall Hi-Ceal compressive forces ranged between 150 and 250 pounds per inch of circumference for deflection of 0.006 to 0.018 inch. The next step was to test thinner wall Hi-Ceals both annealed and as-formed. The force-deflection curves show that annealed 0.010 wall Hi-Ceals met the 100-pound-per-inch-of-circumference requirement. The results of all deflection versus compressive force tests performed on the Hi-Ceals are shown in Appendix II-7 .

The Hi-Ceals test summary in Appendix II-9 shows that several annealed 0.010 wall Hi-Ceals were tested at Chance Vought (S/N 25 through 30) with unacceptable results. In general, the thin-wall seals performed satisfactorily at low pressures when first installed and later at very high pressures. Leakage occurred in the intermediate pressure range. These seals were installed within the groove back-up diameter previously determined as the optimum by High Pressure Engineers.

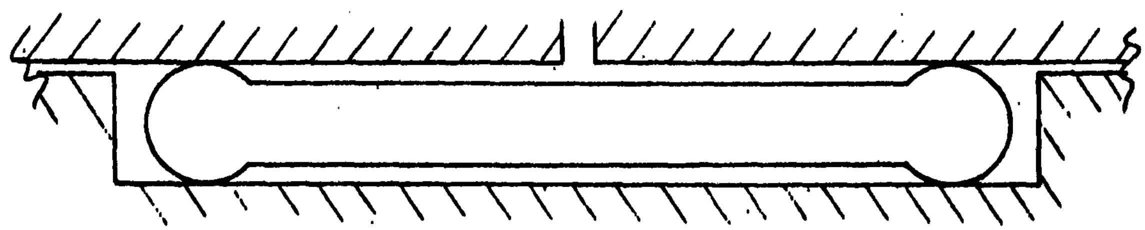
It was felt that seal leakage in the intermediate pressure range was caused by the relatively low initial seal contact force coupled with a further decrease in contact force as the seal expanded diametrically under pressure. When very high (6,000 psi) pressures were applied, the Hi-Ceal cross-section expanded, conforming to the seal cavity with sufficient force to establish a pressure-tight seal. If proof pressure should be applied to the seal upon initial installation, the intermediate pressure leakage might not be detected. This apparently occurred when High Pressure Engineers tested the annealed seal, since they reported satisfactory results in their Phase I and II seal development program. Figure II-2 depicts this type failure.

Tests on annealed Hi-Ceals showed that a seal deflection force somewhat higher than that provided by the annealed seals would be required; and that optimum back-up diameter should be re-defined for thin-wall seals. Specification CVC 2464 was, therefore, amended to include additional testing and development prior to High Pressure Engineers beginning the Phase III qualification testing. High Pressure Engineers was to re-investigate the effects of seal material strength and varying back-up diameters on seal performance, deflection force requirements and seal removability.

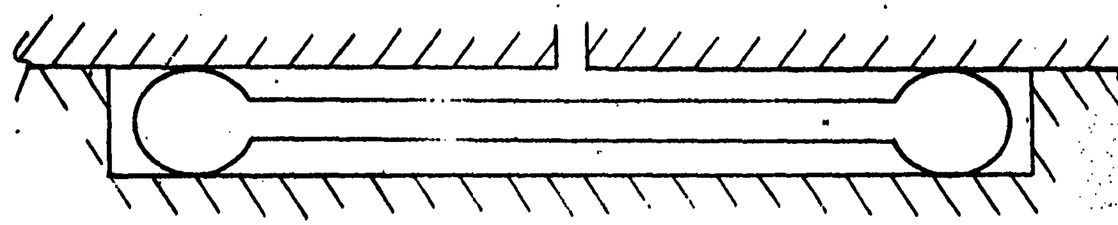
Since annealed seals were not the answer to the excessive installation torque problem because of their unacceptable sealing characteristics, thin-wall seals that were not annealed were tested by both Chance Vought and High Pressure Engineers. It was apparent that a deflection force

FIGURE II-2

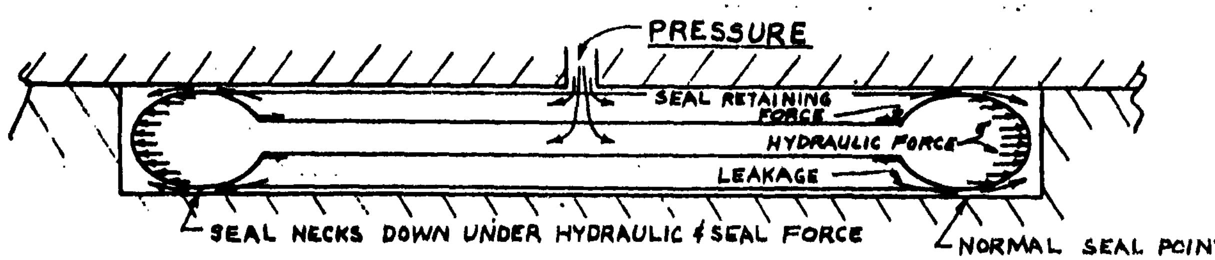
ONE THEORY EXPLAINING LEAKAGE OF ANNEALED HI-CEAL
INSTALLED IN CAVITY WITH EXCESS DIAMETRAL CLEARANCE



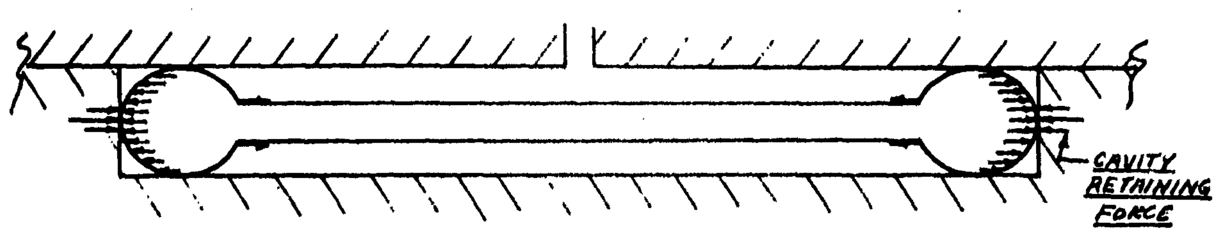
SEAL IN CAVITY BEFORE SQUEEZING



SEAL IN CAVITY AFTER SQUEEZING AND
BEFORE PRESSURIZATION



SEAL UNDER INTERMEDIATE PRESSURE



SEAL AFTER YIELDING UNDER PRESSURE
GAINS DIAMETRAL SUPPORT - REDUCES NECKING

exceeding 100 pounds per inch of circumference would have to be tolerated. Hi-Ceals of 304 material with walls of 0.010 and 0.013 inch were tested. These seals were eliminated from further consideration when several of the Hi-Ceals split during pressure impulse tests. The failure was attributed to fatigue and is discussed in a following section on fatigue.

In a further effort to keep Hi-Ceal compression forces to a desirable value, Chance Vought and High Pressure Engineers undertook studies of thin-wall Hi-Ceals made from AM350 and Inconel X. A discussion of these materials is included in the materials section. An evaluation of the material properties promised to meet seal compression requirements while resisting failures caused by fatigue. Seal compression tests on AM350 and Inconel X with 0.006 wall, and Inconel X with 0.010 wall show force versus compression to be tolerable. Tests on 0.006 wall Inconel X Hi-Ceals (S/N 65, 66, 69 and 70) performed satisfactorily during static tests but leaked badly during pressure impulse test. Leakage was attributed to severe wear on both Hi-Ceal and seats and is discussed in a following section on wear. Tests on the Inconel X, 0.010 wall and AM350, 0.006 wall, also resulted in poor sealing performance as can be seen in the Chance Vought Hi-Ceal test summary in Appendix II-9.

As Hi-Ceal development testing progressed, it became obvious that the 0.015 wall Hi-Ceals of 304 material were outperforming other material and wall thickness combinations. It was concluded that reducing the Hi-Ceal compression force to something less than that of 0.015 wall, 304 material Hi-Ceals could not be accomplished without seriously reducing the sealing and fatigue performance of the Hi-Ceal.

To make the high compressive forces required by the Hi-Ceal more tolerable other means were investigated in an effort to reduce the modular component installation torque. Various platings or coatings on the Hi-Ceal were investigated to reduce friction during component installation and also to reduce wear. A discussion of the platings tried is included in the materials section and also in the section on wear. The results of this test showed that electrolize and teflon coat did reduce installation torque but were considered unsatisfactory for other reasons. Since plating the Hi-Ceals is more concerned with the prevention of wear, it is discussed more thoroughly in that section.

Tests were conducted with various thread lubricants as a means of reducing component installation torque. The test set-up and results of tests are shown in Figure III-28. Tests were first run with MLO-8200 as a lubricant so that other lubricants could be compared with what is actually being used. It is seen that several of the lubricants are vastly superior to MLO-8200 in reducing thread friction. However, use of these lubricants is not recommended because of the contamination that they can introduce into a hydraulic system. If a user of modular components has particular trouble with the torque in some applications, a thread lubricant may be used; but, it shall be the responsibility of the user to determine the effects of the lubricant on his system. In qualifying the modular components, one vendor was allowed to use Silver Goop and another Electrofilm in order to reduce torque. In each case, no trouble was experienced.

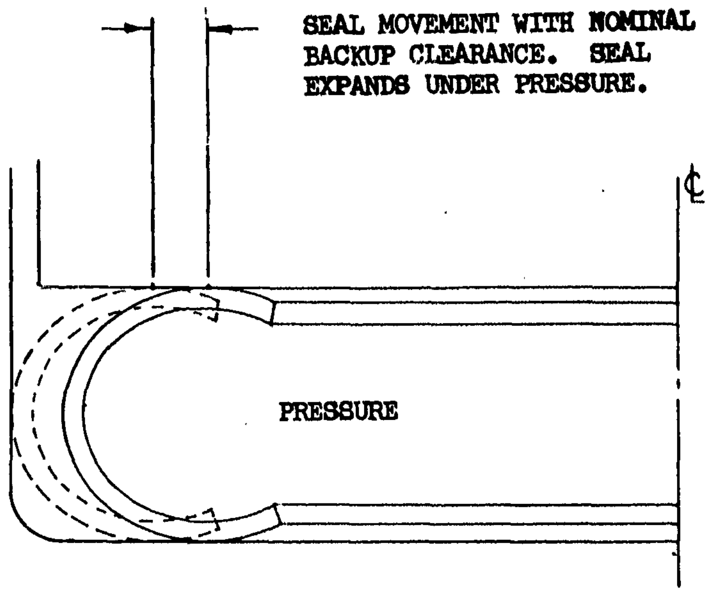
The final results of the torque reduction investigation were that a desirable low component torque could not be obtained without reducing the reliability of the hydraulic system in which components and Hi-Ceals are used. It is concluded that the installation torques for some components take special consideration, but they can be handled. As described in Part II on component development problems, one technique that was used to reduce torque was to pre-set the seals. Pre-setting a seal consists of installing only one seal at a time and bottoming out the modular component so that the Hi-Ceal takes a permanent set.

Wear - Excessive wear of the seals and seats was a problem which was studied throughout the development of the Hi-Ceal. It will be noted in the Hi-Ceal Test Summary (Appendix II-9) that in practically every test it was reported that both seal and seat wear had occurred. Pictures of wear on both the test manifold and Hi-Ceal are shown in the Material Section, Part IV. The wear was, of course, the result of movement between the Hi-Ceal and the cavity seats. Figure II-3 depicts this movement and points out that even with zero back-up clearance movement still takes place. The effect of wear is that it reduces seal life in hydraulic systems subjected to pressure impulses. The wear particles also contaminate the hydraulic system.

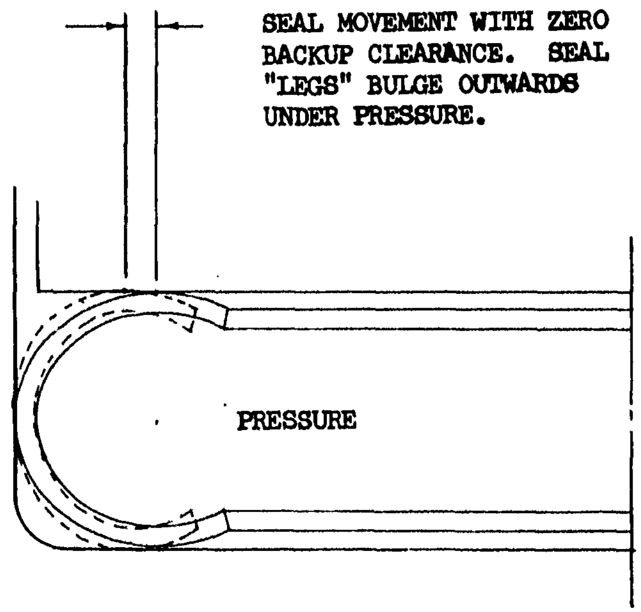
Three types of wear were detected and efforts made to combat each type. The types were galling, abrasion and fretting corrosion.

In coordination discussions with NAA, Inglewood, personnel who are also working on the metallic face seal problem, it was learned that identical wear and fretting corrosion problems have occured with Hi-Ceals and other seal tests. NAA's wear problem was even more extreme because their manifolds are H-11 tool steel with an approximate hardness of Rc 58. Approaches to the present problems involved in the seal development were coordinated with NAA for mutual benefit.

Galling was not as serious a problem as the other two, but it was partly responsible for eliminating the Inconel X material Hi-Ceals. In tests on the Inconel X Hi-Ceals, it was difficult to obtain even a static seal at room temperature. A test with S/N 77 pointed out that cavity seats were damaged during installation -- prior to any pressure tests. The galling is attributed to the high coefficient of friction between the cavity seats and the Hi-Ceal, and the high force required to compress the 0.010 material Hi-Ceal. Galling was not detected on 304 material and AM350 material tests; however, the possibility of it occurring suggested steps be taken to reduce the possibility. Plating the seal with silver plate and nickel plate were tested as means of reducing installation friction. Tests showed that friction was reduced but that the plating work off during impulse testing. The silverplate was later incorporated on the larger size Hi-Ceals. This is discussed in High Pressure Engineers, Inc., qualification report. Since good results were obtained with the unplated 304 material Hi-Ceals, protection against galling was considered minor and was not pursued further.



* CONDITION 1



* CONDITION 2

FIG. II-3

* WITH THE SEAL UNDER PRESSURE BOTH CONDITION 1 AND 2 OCCUR SIMULTANEOUSLY. WITH PRESSURE REDUCED TO ZERO THERE WILL BE A CERTAIN AMOUNT OF SPRING BACK. THE RESULT IS A CIRCUMFERENTIAL WEAR PATTERN ON THE SEATS AND ON THE SEAL.

One of the more troublesome of the wear problems was fretting corrosion. Briefly, fretting corrosion is surface damage that occurs when two solid surfaces are in vibrating contact under high unit loads. Hi-Ceals perform in this manner during pressure impulse cycling as shown in Figures II-4, 5, & 6. In the book, "Deterioration of Materials," by Glenn A. Greathouse, several methods of preventing fretting corrosion were listed. Some of these methods suggested ideas that were tried. Among the methods applicable to the Hi-Ceal development program were: (1) provide lubrication or surface coatings to reduce friction, (2) prevent slippage, (3) increase hardness of one or both surfaces, and (4) reduce the load. All of these methods except Number (4) suggest that a coating such as electrolize, electroless nickel, chrome plate, or silver plate may be required to get the ultimate seal. Much of the testing that was done on seals and seats that were coated was done to prevent or reduce fretting corrosion. Other benefits of the coatings would be to reduce galling and abrasion by reducing friction.

The effects of a soft seal on hard chrome seats was investigated by S/N 96. It is noted that test results are very good during static pressure tests and during early pressure impulse cycling. The apparent sudden failure around 3,700 pic[1] was not accounted for. The upper and lower seats showed very little wear with no evidence of fretting corrosion. The wear on the Hi-Ceal was negligible. One theory is that the chrome-plate seats develop minute cracks which open under pressure allowing a leakage path.

S/N 97 and 98 were electrolized by The Electrolizing Corporation. Briefly, electrolizing is a process which deposits a high chromium alloy on the surface of the basic metal being treated. It was anticipated that the electrolize would increase the pic life of the Hi-Ceal by reducing fretting corrosion. Although evidence of fretting corrosion was very slight after pic, electrolizing increased the surface hardness of the Hi-Ceal to such an extent (approximately Rc 72) that excessive seat wear occurred during installation. This undoubtedly contributed to the early evidence of unacceptable leakage. Finding S/N 98 split after impulse cycling was not surprising since it has already been established that the 0.006 wall is too thin. The 0.006 wall seals were used for these tests because they provided wear data that could be compared to previous tests. It is concluded that electrolizing does help in preventing fretting corrosion, but in order to utilize this process the seal and the seats must be electrolized; or the seats electrolized and the seal remain uncoated.

An electrolized 17-4 plug was used in the test of S/N 119 seal. Electrolizing was done by The Electrolizing Company of Chicago, Illinois. Examination of the plug after 25,000 pressure impulse cycles showed that the electrolized surface had improved the abrasion resistance of the 17-4 and practically eliminated fretting corrosion. In general, it appears that Hi-Ceal impulse life can be increased by using electrolized seats. As for practicability, additional cost and handling required by the process will be a disadvantage.

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[1] pic - "pressure impulse cycles"

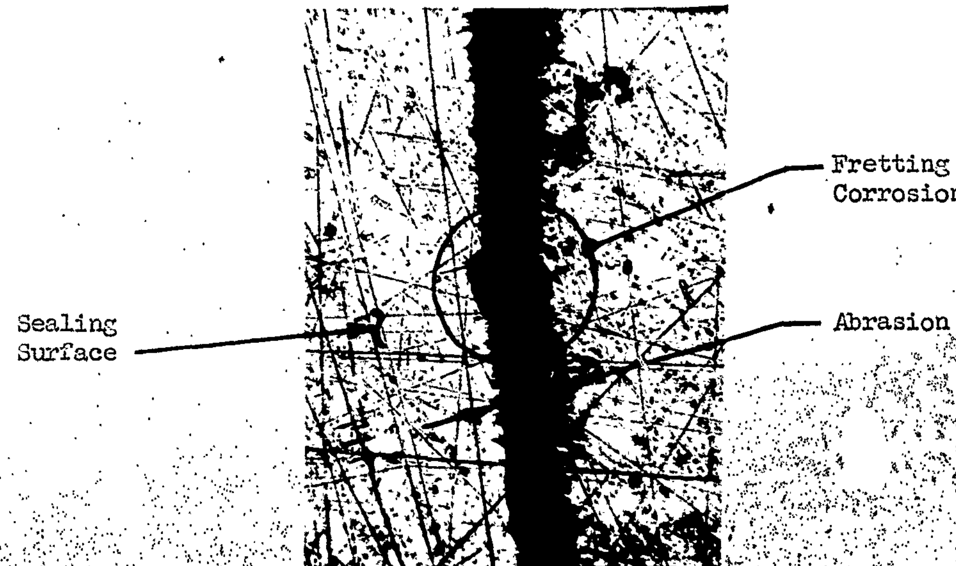


FIGURE II-4

Test Manifold, Lower Wear Plate, 17-4PH Stainless
Heat Treated to 180 ksi. Showing Fretting Corrosion
Area Produced During Test of Hi Ceal.
ML 9501 50X Enlargement

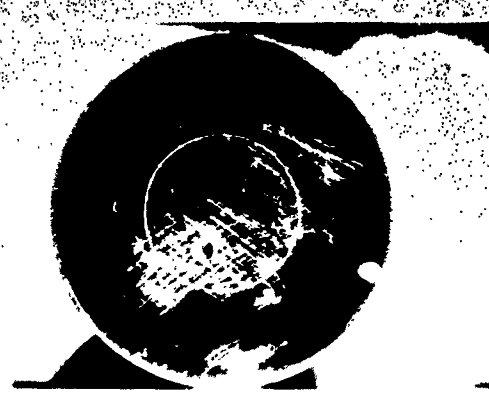


FIGURE II-5

Test Manifold, Lower Wear Plate. 17-4PH Stainless
Heat Treated to 180 ksi. After Pressure Impulse
Cycling of one Hi Ceal
ML [illegible]

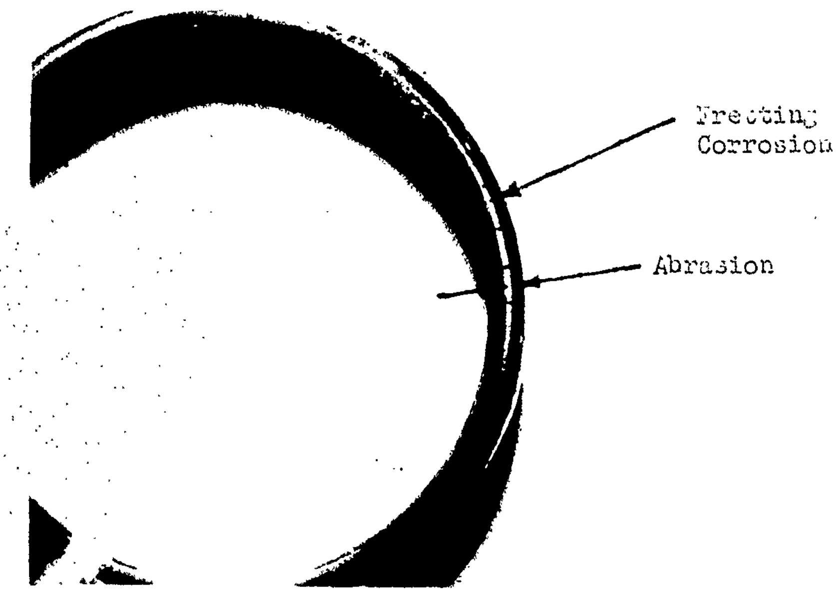


FIGURE II-6

[illegible] Seal after Pressure Impulse Cyclin[illegible]
[illegible]

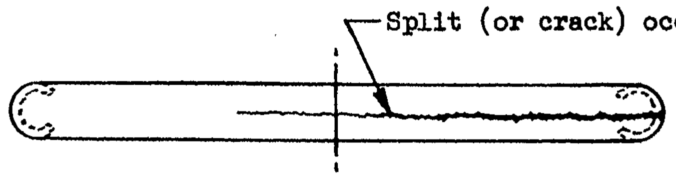
Two Hi-Ceals were teflon coated in an effort to reduce friction, thus improving pressure impulse life. S/N 105B and 128 were coated with a one-coat teflon enamel, which was applied in accordance with process specification CVC 67. The coating is .0007 ± .0001 inch thick and is baked on at 720° ± 20°F. Initial sealing performance was excellent with these seals and impulse life appeared to benefit. However, the teflon was completely worn off the Hi-Ceal where it contacted both seats and where it contacted the diametral back-up during high impulse pressures. It was concluded that teflon coating on Hi-Ceals would not be acceptable in a hydraulic system because of the contamination generated. Tests with nickel plate are discussed in the Material Section, Part IV. Silver plated Hi-Ceals are discussed in High Pressure Engineers, Inc., Hi-Ceal qualification report.

Other steps taken to reduce abrasion and fretting corrosion were to investigate various materials so that those with the greatest resistance to wear could be recommended. While testing Inconel X, AM350 and 304 material Hi-Ceals, the seat materials were varied among 440c, 420, 416 and 17-4PH. All seats were heat treated in the range of Rc 34 to Rc 40. After each test, the seat wear was studied under a 50-power microscope and visual comparisons were made with regard to both abrasion and fretting corrosion resistance. The conclusions reached from these tests were that for modular component sealing applications, 440c exhibited exceptional wear resistance characteristics. The magnitude of its superiority over the other three was somewhat surprising. However, its use is somewhat limited in high pressure application by its high notch sensitivity. The 420, 416 and 17-4PH exhibited similar wear characteristics, with none being superior to the other. Although 440c showed superior wear resistance, it must be concluded that none of the materials were an ideal solution toward eliminating seat wear. For design purposes, any of the above materials are considered acceptable, with such things as notch sensitivity, availability, ease of machining, material cost, etc., playing a role in the final selection.

Figure II-3 not only indicates the manner in which wear takes place, as discussed earlier, but suggests that the way to reduce movement between seat and Hi-Ceal is to use a thicker Hi-Ceal. Since the thinner wall Hi-Ceals of various materials did not perform well, testing eventually converged on the 0.015 inch wall, 304 material Hi-Ceal. This seal clearly outperformed the other Hi-Ceals tested at Chance Vought. This can be seen in the Hi-Ceal Test Summary by noting that a greater number of pressure impulse cycles were being tolerated by the 0.015 wall, 304 Hi-Ceal. Although this Hi-Ceal was declared the optimum configuration with regard to wall and material, Chance Vought was still experiencing a seat wear problem in their test fixture that was disturbing. High Pressure Engineers had reported that in their tests wear was not of the magnitude Chance Vought was experiencing. The main difference in the tests was the type of test fixtures. The wear was satisfactorily reduced and Hi-Ceal performance greatly increased when Chance Vought began to run tests in TL 3618 test fixture in lieu of the TL 3628 test fixture. The newer fixture also reduced a Hi-Ceal fatigue problem. The test fixtures and fatigue problem

are discussed in the next section. Although wear was substantially reduced by switching to the TL 3618 test block, it remains a problem area where improvement may be worthwhile from the standpoint of eliminating contamination particles.

Fatigue - When it was decided that the compressive force of 304 material, 0.015 and 0.019 wall Hi-Ceals was too great, the development program began to test thin-wall Hi-Ceals of different materials. Some thin-wall Hi-Ceals, 0.010 and 0.013 inch, lost their sealing effectiveness and were discovered to have split after as few as 5,700 pressure impulse cycles. The splitting was attributed to the flexing action the seal gets



HI-CEAL FATIGUE FAILURE

during pressure impulse cycling. This flexing eventually work hardens the material causing it to crack or split. The flexing motion is shown in Figure II-3. Splitting also occurred on one AM350 material, 0.006 wall Hi-Ceal. The manner of failure suggested that the walls of the Hi-Ceal be thicker to reduce the flexing. The 0.015 wall, 304 material Hi-Ceal seemed to be the answer. While it was established that the 0.015 wall, 304 material Hi-Ceal had superior sealing ability, Chance Vought detected three failures of this seal by splitting. Since High Pressure Engineers had not experienced this type failure, attention was directed to the test blocks which were the only differences between test methods.

Table II-1 is an accumulation of test data of 304 material, 0.015 wall Hi-Ceal tests performed at Chance Vought. All of these seals are of type zero degree, one-way, 13/16 inch nominal ID (1.00 inch OD). The seal test data is arranged in three groups. Tests performed in TL3628 test fixture with shims are listed in Group 1, tests performed in the TL3628 test fixture without shims are listed in Group 2, and tests performed in the newer TL3618 test fixture are listed in Group 3. See Appendix II-9 for pictures of these test fixtures. The purpose of this table is to present a comparison of initial test conditions to final results.

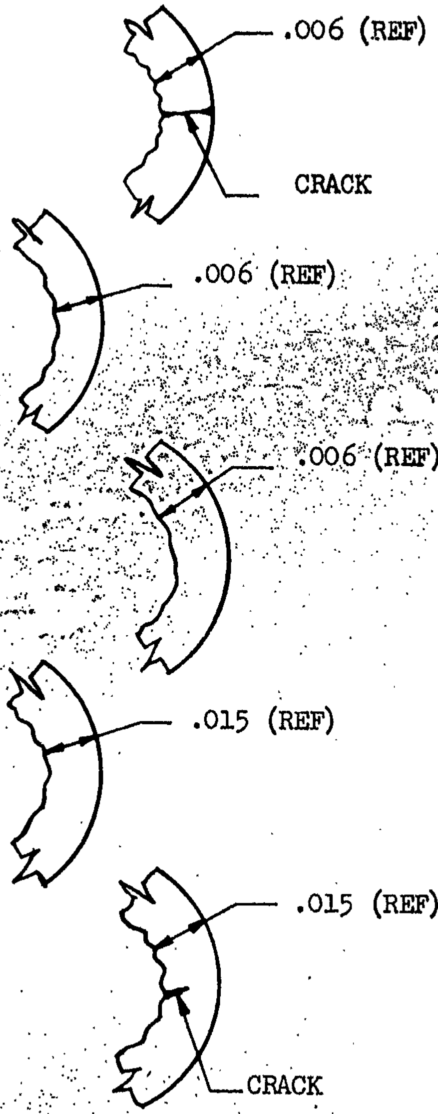
It is noted that two of the Hi-Ceals in Group 1 failed by splitting. In addition, it was determined from a microscopic study of S/N 115 that a crack had started but had not worked its way through. A discussion and photographs of this condition are given in the Materials Section, Part IV, of this report. A sketch is shown in Figure II-7.

Although splitting had been experienced in other Hi-Ceal materials and sizes, it was not expected to be a problem in the 304 material, 0.015 wall Hi-Ceal. The effort to determine the cause of splitting failures

TABLE II-1
TEST DATA

S/N	No. of Shims	Squeeze (in.)	Dia. Backup Clearance (in.)	Seat Matls. Plug	Seat Matls. Base	Pic at 450°F	Max. Leakage Pic cc/hr	Max. Leakage Static cc/hr	Comments
GROUP 1									
120	8	.019	.0017	17-4	416	25,311	13.7	-	Hi-Ceal Split
93	3	.008	.0023	17-4	416	14,890	2.2	-	-
115	4	.008	.0040	440c	440c	12,150	10.0	-	Split started
118	6	.010	.0045	17-4	17-4	7,750	14.4	-	-
119	7	.009	.0056	17-4	420	25,088	12.0	-	Hi-Ceal split
GROUP 2									
149	0	.006	.0035	420	420	32,750	9.5	1.6	-
150	0	.008	.0078	420	420	22,123	14.0	1.6	-
151	0	.020	.0120	17-4	420	21,695	23.0	5.0	-
GROUP 3									
148	0	.018	.0014	416	17-4	50,059	18.0	1	Stuck in cavity
116	0	.008	.0035	416	17-4	35,411	12.6	0	-
117	0	.017	.0120	416	17-4	50,013	1.5	0	-
146	0	.009	.0124	416	17-4	22,123	12.8	0	-
147	0	.018	.0159	416	17-4	49,302	1.0	0	-

MAGNIFICATION OF HI-CEALS (100 X)



S/N 83
AM 350 x .006
Inside of Hi-Ceal rather rough 15,315 pic. Had split in two.

S/N 85
AM 350 x .006
Inside of Hi-Ceal much smoother than S/N 83. 21,750 pic. No cracks detected.

S/N 86
AM 350 x .006
Inside of Hi-Ceal smoother than either S/N 83 or 85. 13,000 pic. No cracks.

S/N 93
304SS x .015
Fairly rough on inside but no cracks. 14,890 pic.

S/N 115
304 SS x .015
Fairly rough on inside. Has beginning of crack but not all the way thru. Had 12,150 pic.

HI-CEAL FAILURES

FIGURE II-7

(No Scale)

As described earlier, one of the quality problems was that of forming wrinkles on the inside of the Hi-Ceal. This problem has been corrected by further improved forming techniques. It is recommended, however, that "wrinkling" be checked on periodically by a user of Hi-Ceals.

It is concluded that the quality of the Hi-Ceal has been improved throughout the program and that production quantities of high quality Hi-Ceals can be obtained from High Pressure Engineers. High Pressure Engineers recently announced that they are procuring automatic machinery for Hi-Ceal production purposes.

Hi-Ceal Materials - A detail discussion of the materials investigated for Hi-Ceals and for other uses in the program is included in Part IV Materials. The following information is a brief summary of what was observed while working with these materials.

Of those considered, type 304 stainless steel was eventually selected as the best material for the Hi-Ceal. The material is easy to form, and strength of materials charts show no loss of strength at 450°F. Tests with different wall thicknesses of this material showed the 0.015 wall to be the optimum configuration. This wall size gave the necessary high contact forces between the Hi-Ceal and the cavity seats, while keeping installation torque to a value that could be tolerated. Compared to other materials, cavity seat wear with the 0.015 wall, 304 material Hi-Ceal was noticeably less.

Tests with all-beta titanium Hi-Ceals have proven this material to be unacceptable. Some of these Hi-Ceals were coated with a moly lubricant, others were coated with anodize, and some were plain. All tests followed the same pattern. Initial sealing was obtained but the Hi-Ceal wore away very rapidly during impulse cycling resulting in early failure (excessive leakage). The anodize and moly-lubricant coatings on the seals quickly wore off and were of no apparent value.

Inconel-X Hi-Ceals were a little more difficult to form than the other materials. Some waviness was detected which made initial sealing difficult. This material also showed a tendency to gall during installation. The material hardness in the as-formed condition was measured as Rc 29. This compares to Rc 19-25 for the 304 in the as-formed condition. Because of the hardness of the material and its galling tendency, excessive cavity seat wear resulted in all tests with Inconel-X Hi-Ceals.

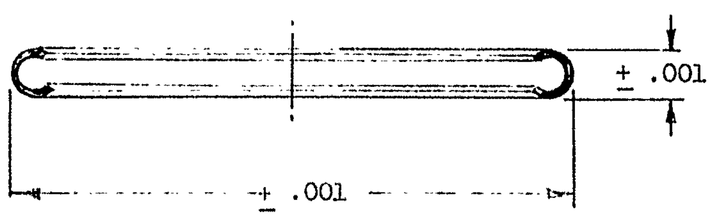
AM350 material exhibited excellent forming properties and the test samples sent to Chance Vought were of high quality. The Hi-Ceals had a measured hardness of Rc 28-30 in the as-formed condition and average hardness of Rc 44 in the aged condition. Hardness of the cavity seats ranged between Rc 35 to Rc 42. Tests results of S/N 65, 66, 69 and 70 indicated that the 0.006 aged seals were too hard and too flexible for this application. Although the hardness of the as-formed AM350 was softer than the aged, excessive seat wear occurred for both the aged and the as-formed AM350 Hi-Ceals. In comparison, AM350 caused greater cavity seat wear than the 304, but less than the Inconel-X.

brings out two factors. Figure II-7 shows various Hi-Ceal cracking conditions. The split starts in the valley of a wrinkle on the inside diameter of the Hi-Ceal. Since the valleys form stress concentration points they are undoubtedly a factor contributing to failure. The wrinkles result from forming the Hi-Ceal and are more evident on some Hi-Ceals than others. This indicates that with proper quality control the wrinkles can be eliminated. High Pressure Engineers are confident that they can prevent wrinkling from occurring by careful forming techniques. The quality of these seals should be periodically checked by users.

The other factor that apparently contributes to the seal splitting is a shortcoming in the TL3628 test fixture design. This fixture is shown in Appendix II-9. In many tests, shims are used under the baseplate to control squeeze on the fixture's elastomeric seal. Although considerable torque is put on the plug to give firm support to the baseplate, indications are that the pressure impulses cause the baseplate to deflect a microscopic amount. This deflection causes a flexing of the Hi-Ceal which eventually results in a fatigue failure of the metal. The failure would be accelerated by the stress concentration points caused by wrinkling. It can be seen from Groups 1, 2 and 3 of Table II-1 that splitting ceased when tests were performed without shims. This fact points out the importance of providing firm Hi-Ceal seats, and the misuse of the shims.

On the basis of number of impulse cycles and leakage rate at the conclusion of each Hi-Ceal test, Group 3 Hi-Ceals outperformed those in Groups 1 and 2. Leakage rates of 12 cc/hr during impulse testing were not considered excessive, since in most applications the seals will not be subjected to the steady 6,000 psi impulsing of development tests. However, static leakage after impulsing is expected to be zero. The improved performance of Group 3 Hi-Ceals is attributed to the TL3618 test fixture which provides firm seats by removing both the baseplate and shims. Hi-Ceals tested in the TL3618 test fixture more closely simulate actual Hi-Ceal useage. Because modular component designers have provided firm seats for the Hi-Ceals and because quality of the Hi-Ceal can be controlled, it is concluded that fatigue failures will not occur in modular systems.

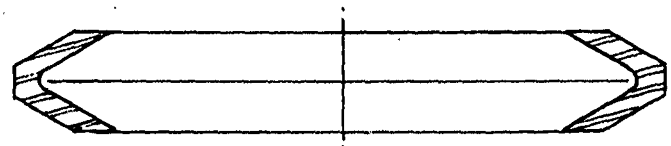
<u>Hi-Ceal Quality</u> - Throughout the development program, High Pressure Engineers continued to improve the quality of the Hi-Ceal. Early in the program, trouble was experienced with holding the close dimensional tolerances of the Hi-Ceal. The final tolerances that were considered necessary and are now being held by High Pressure Engineers are as follows:



Dimensional Tolerance Typical for
Each Type Hi-Ceal

Back-up Seal Tests - While High Pressure Engineers was developing and ultimately qualifying the 0.015 wall, 304 material Hi-Ceal, Chance Vought continued investigation of alternate metallic seals. Tests were performed on the following seals.

Apex Metallic Seal - This seal is made by Servotronics, Incorporated. Chance Vought procured eight of these seals made from 17-7PH stainless steel. Four of the seals were teflon-coated and four were silver-coated. The results of these tests are shown in Table II-2.



Apex Seal

Three observations were made from the limited tests performed:

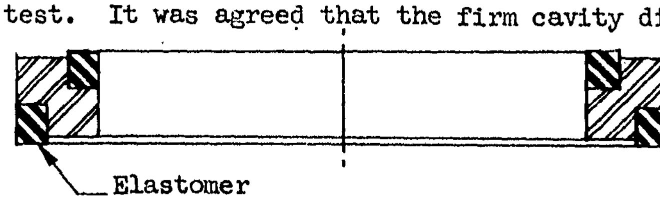
(1) The Apex metallic seal as compared to the Hi-Ceal causes very little cavity seat wear.

(2) Because of its configuration, sealing at high squeeze was unsatisfactory. This indicates that the Apex seal, as it is now designed, will not cover the squeeze range from 0.006 to 0.018 inch which is required by the modular program.

(3) The strength of the Apex seal was not adequate to take impulse pressure tests.

Figure II-8 shows the splitting failures that occurred on four of the eight Apex seals. However, the minimum cavity seat wear and good sealing of the nominal squeeze tests were very encouraging. It is concluded that the Apex seal shows good promise but that further development is required to make it acceptable for modular program use.

A-357 Bonded Elastomer Seal - This seal is made by Precision Rubber Products. Precision Rubber designed and built ten of these seals for Chance Vought to test. It was agreed that the firm cavity dimensions and



A 357 Seal

temperature requirements posed a difficult problem for this type seal. The A-357 seal was tested with unacceptable results. The seals would not pass the static pressure tests at 450°F. When removed the elastomer was found to have severely extruded.

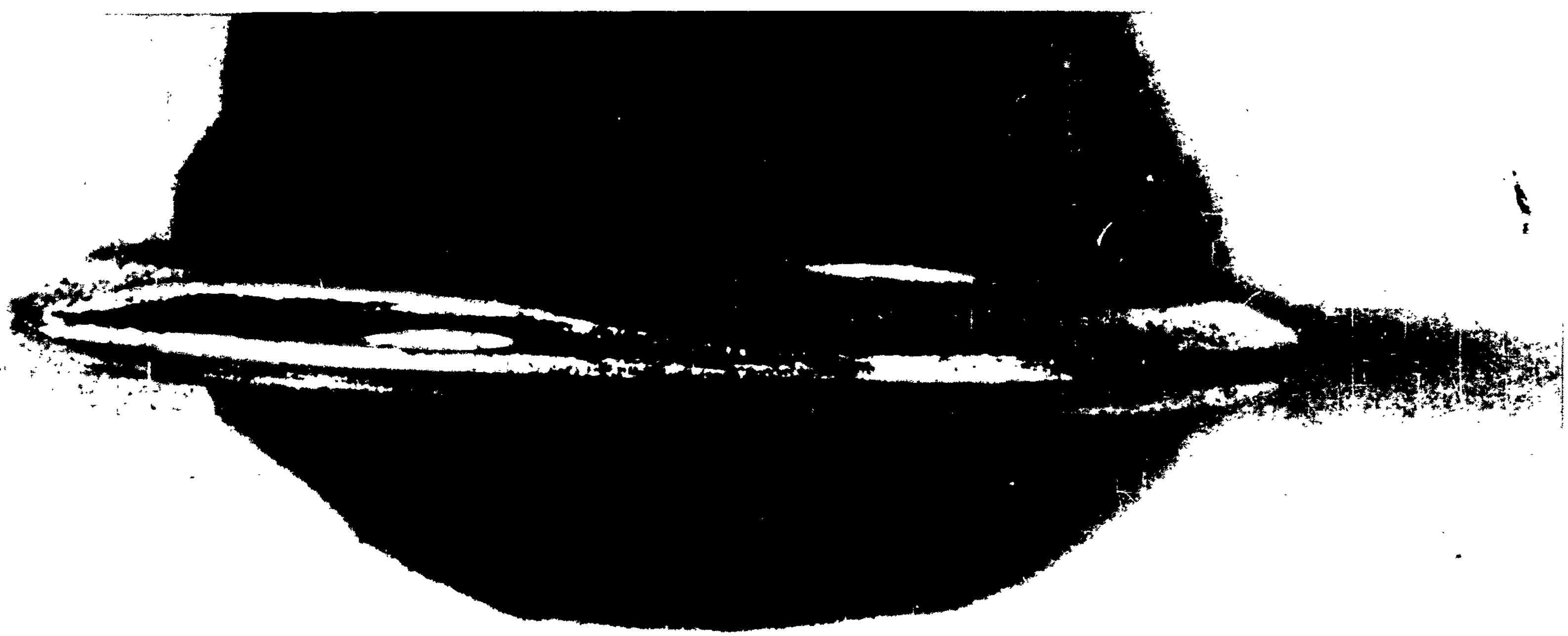
TABLE II-2

SUMMARY OF APEX METALLIC SEAL TESTS

S/N	SEAL MEASUREMENT O.D. - Inches	SEAL MEASUREMENT Depth - Inches	CAVITY MEASUREMENT I.D. Inches	CAVITY MEASUREMENT Depth-Inches	Test Details
1 17-7 Silver Coated	1.0022	(Initial) .0948 - .0950	1.0049	.0855	Satisfactorily sustained proof pressure at room temperature and at 450°F. Seal split at 13,461 p.i.c. around approx. 45° of circumference. Plug and base had thin wear line but no evidence of fretting corrosion pits. Seal had some pits visible with naked eye. Leakage rate up to failure was negligible.
	1.0039	(Post Test) .0920 - .0925	TL 3628-3 Manifold (440 plug, 416 base) (Finish: 6-10 RMS) (Hardness: Rc 37)		
	Squeeze: .0093 min.; .0095 max. Diameter Clearance: .0027				
2 17-7 Silver Coated	1.0018	(Initial) .0944 - .0948	1.0168	.0740	Satisfactorily sustained proof pressure at room temperature and at 450°F. Removed seal after 2,402 p.i.c. due to excessive leakage. Squeeze and backup clearance probably excessive.
	1.0125	(Post Test) .0750 - .0755	TL 3628-3 Manifold (17-4 plug & base) Finish: 6-10 RMS (Hardness: Rc 43 plug Rc 34 base		
	Squeeze: .0204 min.; .0208 max. Diameter Clearance: .0150				
3 17-7 Silver Coated	1.0022	(Initial) .0947 - .0950	1.0042	.0850	No leakage during proof pressure at room temperature or 450°F. Seal sustained 50,000 p.i.c. with negligible leakage. Silver coat was worn from the seal but there was no evidence of fretting corrosion. Base and plug wear was minor.
	1.0045	(Post Test) .0850	TL 3649-1 plug (416 plug; 17-4 base) (Finish: 6 RMS) (Hardness: Rc 34)		
	Squeeze: .0097 min.; .0100 max. Diameter Clearance: .0020				
4 17-7 Teflon Coated	1.0005-1.0010	(Initial) .0948 - .0952	1.0040	.0855	Zero leakage during proof pressure at room temperature and 450°F. Seal split at 8,810 p.i.c. Teflon separated from metal in spots. Wear was negligible on plug and base.
	1.0035	(Post Test) .0865 - .0870	TL 3628-3 manifold (17-4 plug; base) (Finish: 8 RMS) (Hardness: Rc 43)		
	Squeeze: .0093 min.; .0097 max. Diameter Clearance: .0030 min.; .0035 max.				

SUMMARY OF APEX METALLIC SEAL TESTS (Continued)

S/N	SEAL MEASUREMENT O.D. Inches	SEAL MEASUREMENT Depth Inches	CAVITY MEASUREMENT I.D. Inches	CAVITY MEASUREMENT Depth Inches	Test Details
5 17-7 Teflon Coated	(Initial) 1.0000 (Post Test - NONE) Squeeze: .010 Dia. Clearance: .0029	.095	1.0029 TL 3649 plug (416 plug; 17-4 base) (Finish: 6-10 RMS) (Hardness: Rc 34)	.0850	Seal split after16,051 p.i.c. Teflon badly worn from seal. Leakage data not available.
6 17-7 Teflon Coated	(Initial) 1.000 (Post Test - NONE) Squeeze: .0105 Dia. Clearance: .0049	.096	1.0045 TL 3628-3 manifold (17-4 plug; 416 base) (Finish: 6-10 RMS) (Hardness: Rc 37	.0855	Seal split after 6,119 p.i.c. Leakage slight up to failure.
7 17-7 Silver Coated	(Initial) 1.0015 - 1.0020 (Post Test - NONE) Squeeze: .0155 Dia. Clearance: .0057 min. .0062 max.	.0950	1.0077 TL 3649 Plug (416 plug; 17-4 base) (Finish: 6-10 RMS) (Hardness: Rc 34)	.0795	Successfully sustained 50,000 p.i.c. Very slight leakage. Wear line thin but could be felt with fingernail. No fretting corrosion.
8 17-7 PH Teflon Coated	(Initial) 1.0005 - .9990 (Post Test) No measurements - seal split. Squeeze: .016 nom. Dia. Clearance: .0018	.0942 nom.	1.0023 TL 3628-3 manifold (17-4 plug and base) (Finish: 6-10 RMS) (Hardness: Rc 43 plug Rc 34 base)	.0780	No leakage during proof pressure at room temperature or 450°F. Seal split after 13,369 p.i.c.



[illegible] II-8 [illegible] /N 1-17-7 [illegible] WITH [illegible] COATING - [illegible] AT 13,461 [illegible]

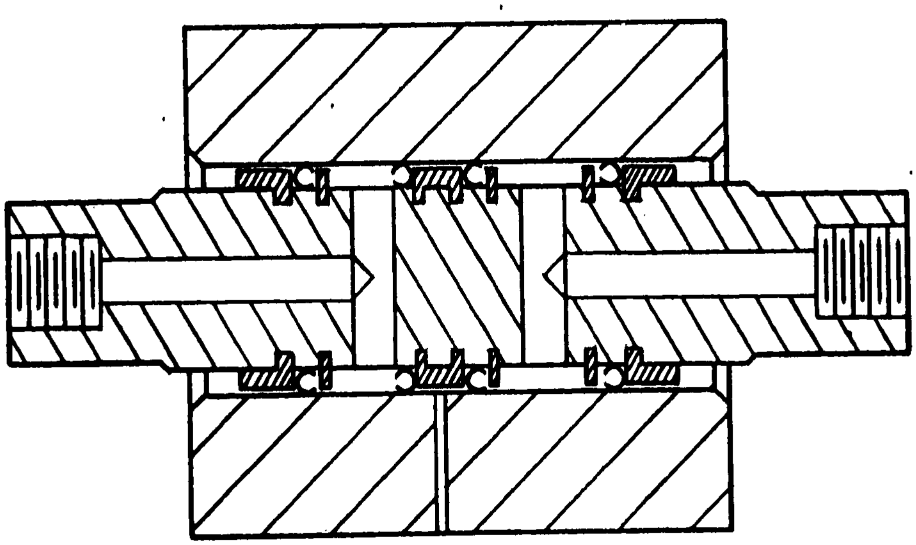
In addition to the suitability tests on the Apex and the A-357 seals, tests were performed with the 0.017 and 0.019 wall, 304 material Hi-Ceals. Test results show that these seals did not out-perform the 0.015 wall, 304 material Hi-Ceal. The reduced performance of the thick-wall Hi-Ceals was attributed to the very high force required to squeeze these seals. The high squeeze force, in conjunction with friction between seals and Hi-Ceals, apparently caused the Hi-Ceal to gall and distort under installation torque. This sometimes prevents even initial sealing as evidenced by S/N 124 and S/N 124A tests. In an effort to reduce the friction and the resulting galling and distortion, S/N 128 was teflon-coated and tested. The test shows that a low coefficient of friction between the Hi-Ceal and seats greatly improves performance of the seals; however, the teflon contaminates the system as it wears off presenting an undesirable condition. A search has been made for a more rugged coating without successful results. The high installation torque required by the thicker wall Hi-Ceals and the fact that they have not exhibited superior sealing characteristics eliminated them from further consideration.

7. Selection of a Radial Wedge Seal

The study of modular component configurations indicated that a radial seal would be required in modular cartridge, 4-way selector valve designs. To stimulate interest in this type metallic seal, Chance Vought Specification M.H.2 was sent to various companies. This specification is discussed earlier in the report and is included as Appendix II-2. Promising seal proposals were submitted by Bendix-Pacific, High Pressure Engineers, Cadillac Gage, and Vinson. Feasibility study contracts were awarded to Bendix-Pacific and High Pressure Engineers, with Chance Vought Specification CVC 2461 covering the detail requirements. This specification is included in Appendix II-5. Due to the size of the Cadillac Gage design and possible dis-assembly problems with the Vinson design, these approaches were not considered feasible for this application.

8. Results of Feasibility Studies

The final report from High Pressure Engineers covering their radial seal feasibility contract stated that radial (90°groove) seals having a 0.015 wall thickness, silver-plated, in the "as formed" condition, met the specification requirements with the exception of a weeping action which started after 125,000 impulse cycles. These seals passed the proof pressure test with no leakage after 200,000 pressure impulse cycles. The weeping action amounted to approximately one drop of fluid in 10 minutes of impulse testing. The axial force required to insert four 1.250 O.D. radial seals into the test fixture was approximately 155 pounds. A sketch of the radial seal test fixture is shown below. The back-up ring configurations shown were determined, after tests, to minimize seal motion under pressure impulsing. Because the installation forces for this seal would be excessive for a modular component, development of this configuration of the Hi-Ceal was discontinued.



Radial Hi-Ceal Test Fixture (Four Seals)

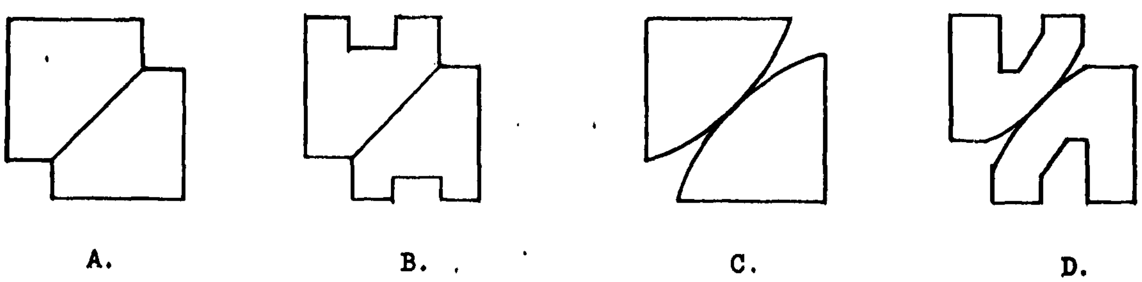
The Bendix-Pacific radial wedge seal successfully completed feasibility studies. The results of these studies are included in the following Bendix-Pacific reports: Report No. 7-778, Report No. 7-778A, and Report No. 7-778B. The manner in which the metallic wedge seals are to be used in the modular program is shown in Figure II-9.

On the basis of the feasibility study, Bendix-Pacific was awarded an additional contract to fully qualify the optimum wedge seal design to modular program requirements specified in CVC 2461. The wedge seal was successfully qualified in the two sizes specified by CVC 2461. The results of qualification tests are included in Bendix-Pacific reports 7-1120 and 7-1224.

9. Development of the Bendix Radial Wedge Seal

Although complete development details can be found in the reports mentioned above, a summary of the wedge seal development is included here for convenience. It will be noted from Specification CVC 2461 that the wedge seal program was divided into three phases. Phase I was concerned with proving the wedge seal concept by performing feasibility tests and studies. Phase II required that an optimum wedge seal design be qualified for use in the Class "A," 4-way, 3-position, solenoid-operated cartridge valve. Phase III required that an optimum wedge seal be qualified for use in the Class "C," 4-way, 3-position, solenoid-operated cartridge valve.

During Phase I, four different wedge configurations were studied. The seals were subjected to hydraulic pressure testing with configuration "C"



BENDIX SEAL CONFIGURATIONS

seal proving most satisfactory in meeting requirements of Specification CVC 2461. All hydraulic testing was performed in test fixtures which approximated the actual application as shown in Figure II-9 . One difference was that the feasibility study test fixtures accommodated only four sets of rings. Although sealing ability was very good, the torque required on the collar to effect zero leakage was too great.

In order to reduce torque requirements from the prohibitive 300 ft./lb. torque required by the original configuration, the wedge seal was redesigned. The design changes consisted of the following items:

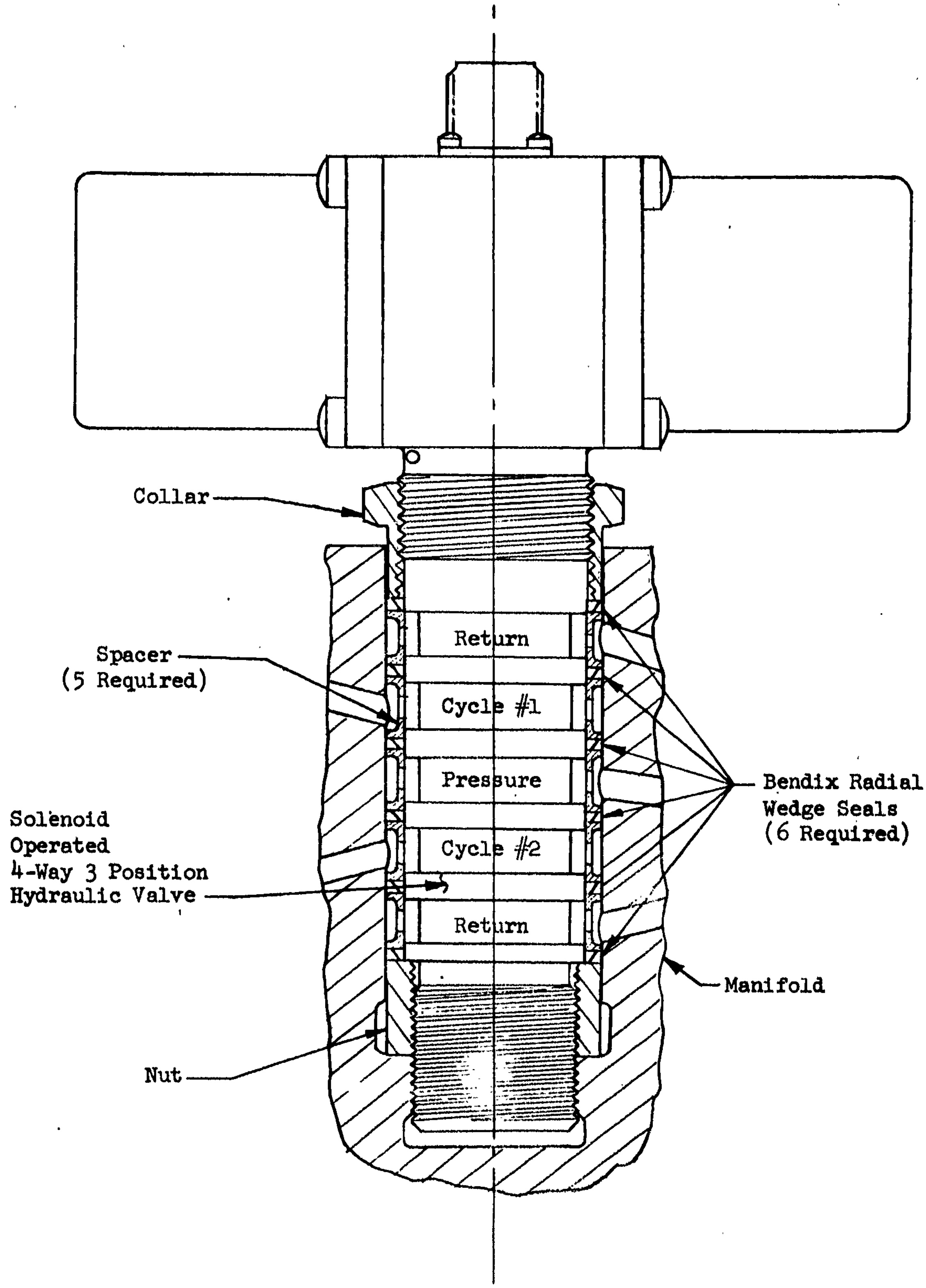


FIGURE II-9
APPLICATION OF BENDIX RADIAL
WEDGE SEALS

a. Reduction of longitudinal width of each wedge from its original width of 0.140 ± 0.005 to 0.055 inch.

b. Reduction of wedge angle from 45° to 37°. This change did not affect the self-releasing feature when dis-assembling.

c. A chamfered wedge on the outer wedge and a radiused surface on the inner wedge. This is to allow some measure of misalignment without affecting sealing at the mating surfaces of the wedges.

d. Change of material from 17-4PH at Rc 35 to 440c stainless steel heat treated to Rc 55-60.

e. Elimination of silver-plating.

It was also pointed out that the threads on which the collar rotates must have a lubricant. This was necessary to reduce friction in the threads, thereby allowing a greater percentage of the applied torque to be transmitted to the seals. It was found that the wedge seal sealing torque increased almost as a direct function of increased thread coefficient of friction. Bendix used Electro-film No. 2006 on the threads of both the body and the collar. The Electro-film No. 2006 was deposited to a thickness of 0.0002 to 0.0004 inch and is capable of withstanding sustained temperatures up to 750°F. After approximately ten disassemblies of the test unit, it was necessary to renew the electro-film deposit on the threads of the collar.

The above changes were successful in reducing installation torque from 300 to 70 ft./lbs. while maintaining zero to very low leakage (4 to 5 drops per minute). Because of this success, Bendix was awarded the contract for Phase II wedge seal development and qualification as described in CVC 2461. Before starting qualification, Bendix was to investigate the possibility of changing the hardness of the wedge seals from Rc 55-60 to Rc 34 to obviate the possibility of damage or wear to manifold sealing surfaces.

The Phase II seals were fabricated from type 17-4PH corrosion-resistant steel. These seals had a measured hardness of Rc 34. Seal design and size were identical to the final seal configuration developed under the Phase I feasibility study. Torque and leakage tests were performed upon a seal assembly containing four pairs of sealing rings. These test results indicated satisfactory sealing with the Rc 34 seals at 60 ft./lbs. of torque versus 70 ft./lbs. for the Rc 55-60 seals developed under Phase I.

Subsequent tests called for comparison of the torques required for sealing up to six pairs of sealing rings of both hardness ranges installed upon a dummy modular valve. Both the soft seal and the hard seal were tested to determine the torque required to effect zero leakage with 2, 3, 4, 5, and 6 pairs of sealing rings when subjected to a static pressure of 6,000 psi. The soft seals were fabricated from type 17-4PH corrosion-resistant steel and had a measured hardness of Rc 34. The hard seals were fabricated from type 440c corrosion-resistant steel and had a measured hardness of Rc 58.

A plot of torque versus pairs of seals showed an almost straight-line relationship between the number of seals and the torque required for sealing. Six pairs of seals required 130 ft./lbs. of torque, approximately three times the torque required for two pairs of seals. There was no practical difference between the torque required for the soft seals and that required for the hard seals. However, the reported torque of 130 ft./lbs. for six pairs of rings sized for the Class "A" selector valve was not considered ideal and work was undertaken to reduce torque below 100 ft./lbs. Four possibilities to reduce torque were noted; namely, (a) reduce seal wedge angle, (b) change seal material, (c) hold tighter cavity tolerances, and (d) decrease seal cross-section.

Development tests showed that reducing seal wedge angle and changing seal material did not result in a significant torque reduction. A complete set of seals was fabricated and tested which had the seal cross-section axial height reduced by one third (from 0.06 to 0.04). Some slight reduction in torque was realized. However, this reduction in torque could not be justified when compared to the additional difficulty and expense of manufacturing the thinner section to the close dimensions required. Seals with the cross-section radial width reduced were not fabricated since a reduction in seal radial width would lessen the fluid flow area around the circumference of the valve to an unacceptable level.

The only change which showed a worthwhile reduction in torque was to hold tighter seal cavity tolerances. The allowable tolerance was previously 0.002 on the valve diameter and 0.002 on the manifold bore diameter. Torque tests indicated that satisfactory sealing would be obtained with a maximum of 95 ft./lbs. if the tolerances were reduced to 0.001. A manifold bore diameter of 1.4600 to 1.4611 and a valve diameter of 1.2120 to 1.2109 was recommended by Bendix and approved by Chance Vought for the Class "A" 4-way selector valve.

After solving the torque problem by reducing valve and housing bore diameter tolerances, it was decided to qualify 440c seals but to reduce hardness to Rc 34. Bendix completed fabrication of test blocks and seals that were to be used for qualification. Upon receiving detail drawings of the test blocks and seals, it was noted that the test blocks were not of the thin-wall configuration required for qualification. A drawing was sent to Bendix showing changes required to make their test block acceptable. This change required Bendix to reduce their test housing wall thickness to 0.12 inch. This change was completed and qualification testing started. A drawing of the Class "A" wedge seals is shown in Figure II-10.

The Class "A" radial wedge seal satisfactorily completed qualification tests in accordance with specification CVC 2461. Pictures of the qualified parts and test fixture are shown by Figures II-11 and II-12. The seals are tested in two units which were fabricated to provide minimum clearance tolerances between the various sealing surfaces. A set of six wedge seals installed in the minimum clearance unit provided zero leakage sealing at a hydraulic pressure of 6,000 psi for a period of 5 minutes with a loading torque of 95 ft./lbs. applied to the collar. A set of six wedge seals installed in the maximum clearance unit provided zero leakage sealing at a

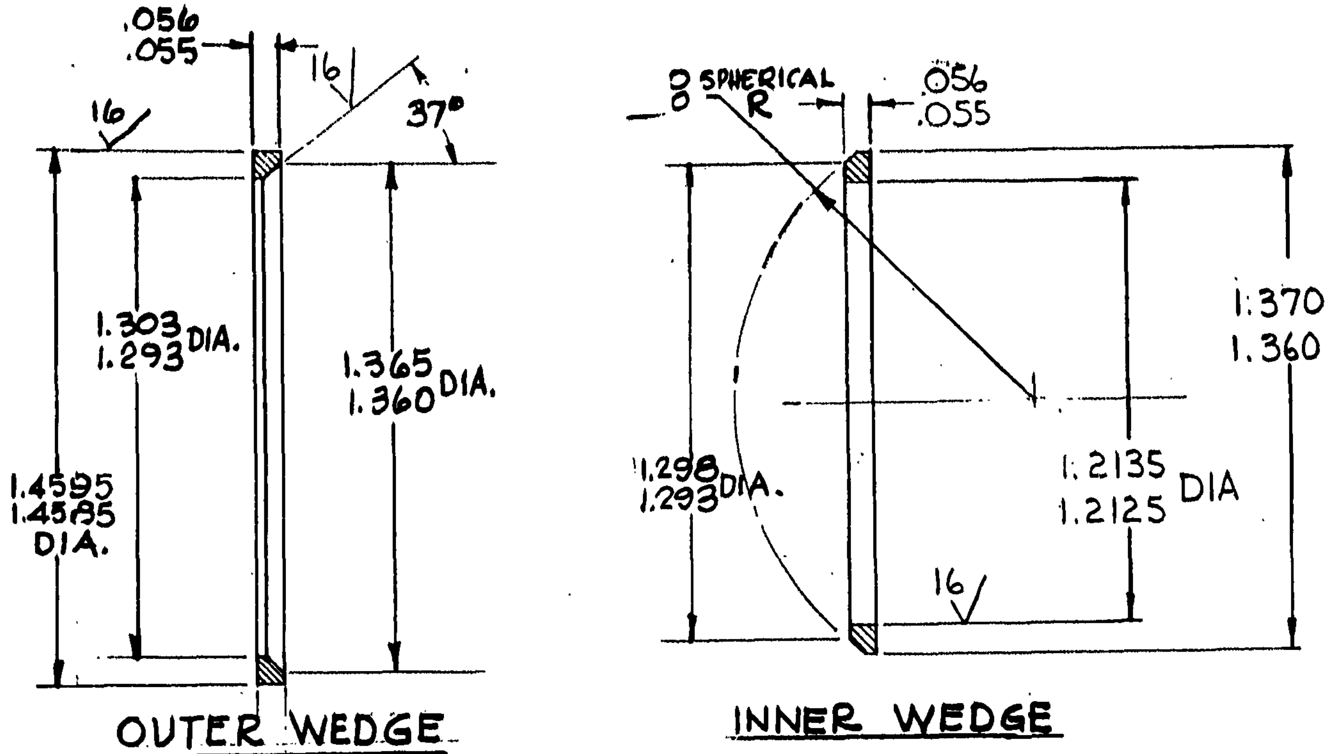


FIG II-10
2X SIZE

hydraulic pressure of 6,000 psi for a period of 5 minutes with a loading torque of 100 ft./lbs. applied to the collar.

Both test units demonstrated their ability to withstand repeated disassembly and re-assembly without loss of sealing capacity. The only leakage problems encountered during qualification testing occurred sometime after 125,000 impulse cycles of 6,000 psi had been imposed upon the seals.

Both test units were subjected to 200,000 impulse pressure cycles at temperatures ranging from -65° to +450°F. The maximum clearance assembly conformed to all requirements of Chance Vought Specification CVC 2461. However, the minimum clearance unit leaked during the 10,000 psi burst pressure test. It was determined that the various sealing surfaces of this unit had been damaged due to fretting corrosion (see figure II-11).

Bendix recommended that the material hardness of the wedge seals be increased from Rc 30-35 to Rc 55-60. This improvement should significantly reduce the fretting corrosion problem. Chance Vought approved this recommendation and Bendix used wedge seals of this hardness for the Class "C" qualification tests.

Preliminary testing on the Class "C" wedge seals to determine torque requirements for zero leakage sealing at 6,000 psi hydraulic pressure were unsuccessful. The maximum clearance unit leaked profusely past all seals when torqued to 100 ft./lbs. Leakage flow started at approximately 3,000 psi at the comparatively low rate of approximately 1cc per minute. The leakage rate increased with pressure. At approximately 5,000 psi, a full stream flow occurred past all seals. The minimum clearance unit was then assembled using the same seals as used in the maximum clearance unit. A torque of 100 ft./lbs. was applied and each seal tested individually at 6,000 psi. All six seals leaked profusely at 6,000 psi. It was noted again that leakage rate was a function of pressure. Little, if any, leakage occurred below 3,000 psi. The torque was increased to 140 ft./lbs.

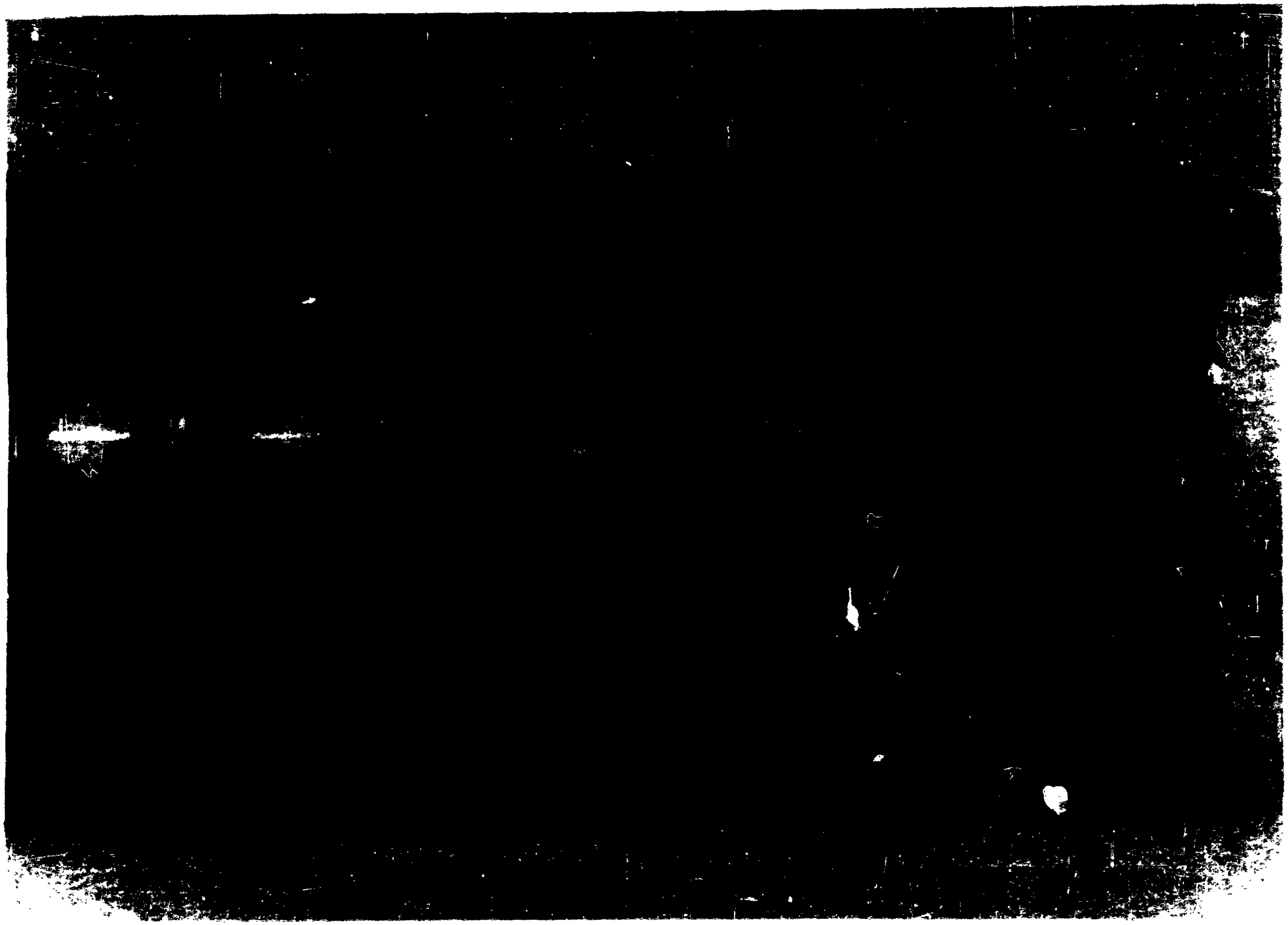


FIGURE II-11 WEAR ON CLASS "A" BENDIX WEDGE SEALS AFTER QUALIFICATION TESTING

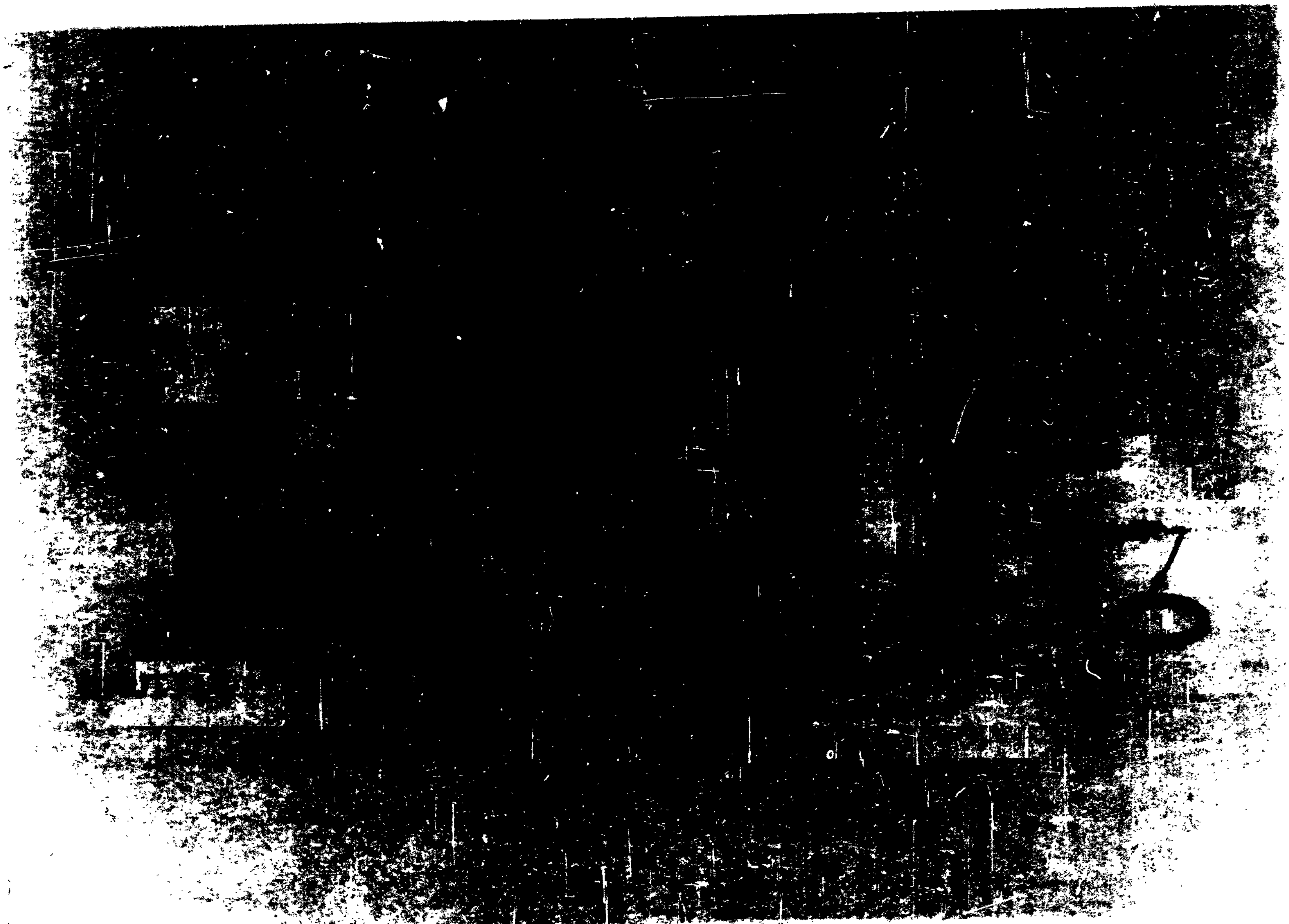


FIGURE II-12 TEST FIXTURE FOR BENDIX WEDGE SEALS

and the test repeated. Leakage was less, but still unacceptable. A dial indicator was set up to measure deflection of the cylinder wall when an internal pressure of 6,000 psi was applied. Bendix learned that the 0.125 wall test housing being used for the Class "C" wedge seals was expanding in an elliptical manner with as high as 0.006 diametrical deflection in one plane as compared to approximately 0.003 diametrical deflection in the other plane. These high test housing wall deflections present extremely adverse conditions for proper functioning of the radial seals.

Bendix decided that an increase in the test housing wall thickness for the Class "C" valve would be required to reduce wall deflections to an acceptable value. Calculations were made by both CVC and Bendix based on the assumption that Class "C" test housing deflections could not exceed deflections that were present in the Class "A" test fixture. Based on these assumptions, it was determined that the fixture wall thickness must be increased from 0.125 to 0.230 inch.

Qualification of the Class "C" wedge seals continued with new test fixtures made from 17-4PH (Rc 36-42) material with wall thicknesses of 0.288 inch. The Class "C" wedge seals successfully met the requirements of Specification CVC 2461. As with the Class "A" seals, some leakage was evident during proof and burst pressure tests after 200,000 pressure impulse cycles. Again, fretting corrosion was evident but had been significantly reduced over that in Class "A" tests by increasing hardness of the wedge seals to Rc 55-60. Figure II-13 & 14 may be compared with that shown in Figure II-11 to note difference in fretting corrosion.

Based on qualification test results, it is concluded that Bendix radial wedge seals can be successfully used with the Class "A," Class "B" and Class "C" modular hydraulic, 4-way, 3-position, cartridge-type selector valves. This conclusion is further substantiated by their successful use in the qualification testing of the 4-way, 3-position cartridge selector valves. In using these seals, the following installation data should be adhered to:

1. The diametral sealing surface finish of the body and shuttle should be 16 RMS maximum.

2. Recommended torque is 90 to 110 ft./lbs. for six seal pairs in series. Recommended torque for fewer seals in series could be reduced in proportion to the number of seals to approximately 25-30 ft./lbs. for two pairs in series.

3. Minimum housing wall thickness for the Class "A" seals is recommended at 0.120 inch. Minimum housing wall thickness for Class "C" seals is recommended at 0.218 inch. These wall thickness recommendations apply only to the sealed area. The required extention on each side of the seal for this thickness is not known at this time.

4. A thread lubricant should be used on the seal loading parts to minimize torque and to make the sealing torque as consistent as possible. Thread lubricant used in the test units was Electro-film No. 2006 coated to 0.0002 to 0.0004 thickness.

5. Normal handling precautions for any precision parts should be observed in order that seals or other related parts are not scratched or damaged prior to, or during, installation or removal.

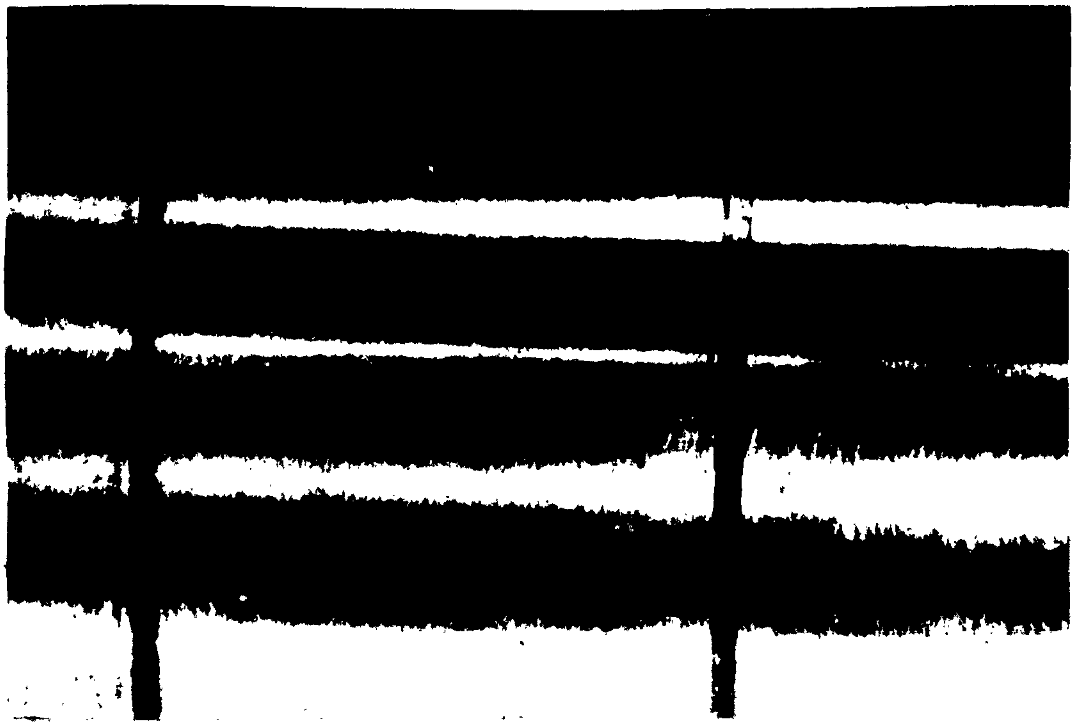


Figure 13 - Wear rings on shuttle of Class C Bendix Wedge Seal Test Fixture after Qualification Testing. Note Fretting Corrosion.

Figure 14 - Wear marks on outer ring of Class C Bendix Wedge Seal after Qualification Testing.

C. Conclusions:

High Pressure Engineers' Hi-Ceal and Bendix Pacifics' radial wedge were developed, qualified, and deemed ready for use with modular components in Type III hydraulic systems. The readiness of the metallic seals is based upon the following events:

(1) During the development program all problems associated with modular component metallic seals were thoroughly investigated and satisfactory conclusions obtained.

(2) Both High Pressure Engineers and Bendix Pacific successfully qualified their metallic seals to Chance Vought specifications CVC 2461 and CVC 2464 respectively.

(3) Modular component vendors developed and completely qualified their components using Hi-Ceals and Bendix wedge seals with satisfactory results.

Although satisfactory seals are available for modular hydraulic usage the development programs brought to light several problems that the Hi-Ceal and Bendix wedge seal did not ideally satisfy. These problems are the excessive installation torque of some components caused by large Hi-Ceal deflection forces, noticeable wear caused by pressure impulse cycles, inability to reuse the Hi-Ceal without certain restrictions, and the undesirable weight that results from thick wall manifolds which are necessary to keep wall deflections nil when using the Class C Bendix wedge seals.

III. PACKAGE DEVELOPMENT

A. Introduction

The design approach to the packaging of several hydraulic components in one housing, that has been developed for the modular hydraulics program, is new and somewhat unique. Packaging techniques have been used for many years in isolated applications but such designs were especially tailored for specific requirements and any attempt to standardize the components integrated into these packages for possible other uses, was not feasible or expedient. The uniqueness of the modular approach is that a series of standard hydraulic components have been developed which readily lend themselves to packaging. This approach is depicted in Figure III-1.

B. Package Design

Packaging two or more hydraulic components into a single manifold has been appealing because of the following advantages:

(1) Reduction in space
(2) Reduction in weight
(3) Reduction in number of system parts
(4) Less leak points
(5) Less maintenance
(6) Easier to install
(7) Reduction in spare parts, stocking of items, paperwork, etc.

However, because of a number of disadvantages, packaging aircraft hydraulic systems has been limited to very selective applications. The disadvantages of packaging are:

(1) Defies standardization
(2) Requires new design, development, testing and qualification
(3) Requires more time from concept to production
(4) Inflexible design -- cannot be changed readily
(5) Expensive to fabricate in small quantities
(6) Limitation on choice for location
(7) Greater risk due to complications of new development
(8) Logistics problem created by non-standard parts

In developing a component design that would lend itself to packaging Chance Vought was concerned with maintaining the inherent advantages of packaging mentioned above, while either eliminating or minimizing the disadvantages. The two most feasible designs for the modular components were determined to be the drop-in and the screw-in types. A comparison of the two types showed that the greatest benefit was derived from the screw-in configuration. As a result of early investigations, all components were designed as self-contained screw-in units. This type component provides for the retention of all the inherent advantages of packaging and has the following effect on the above-mentioned disadvantages:

(1) The modular components are standard, which allows some degree of standardization of packaging.

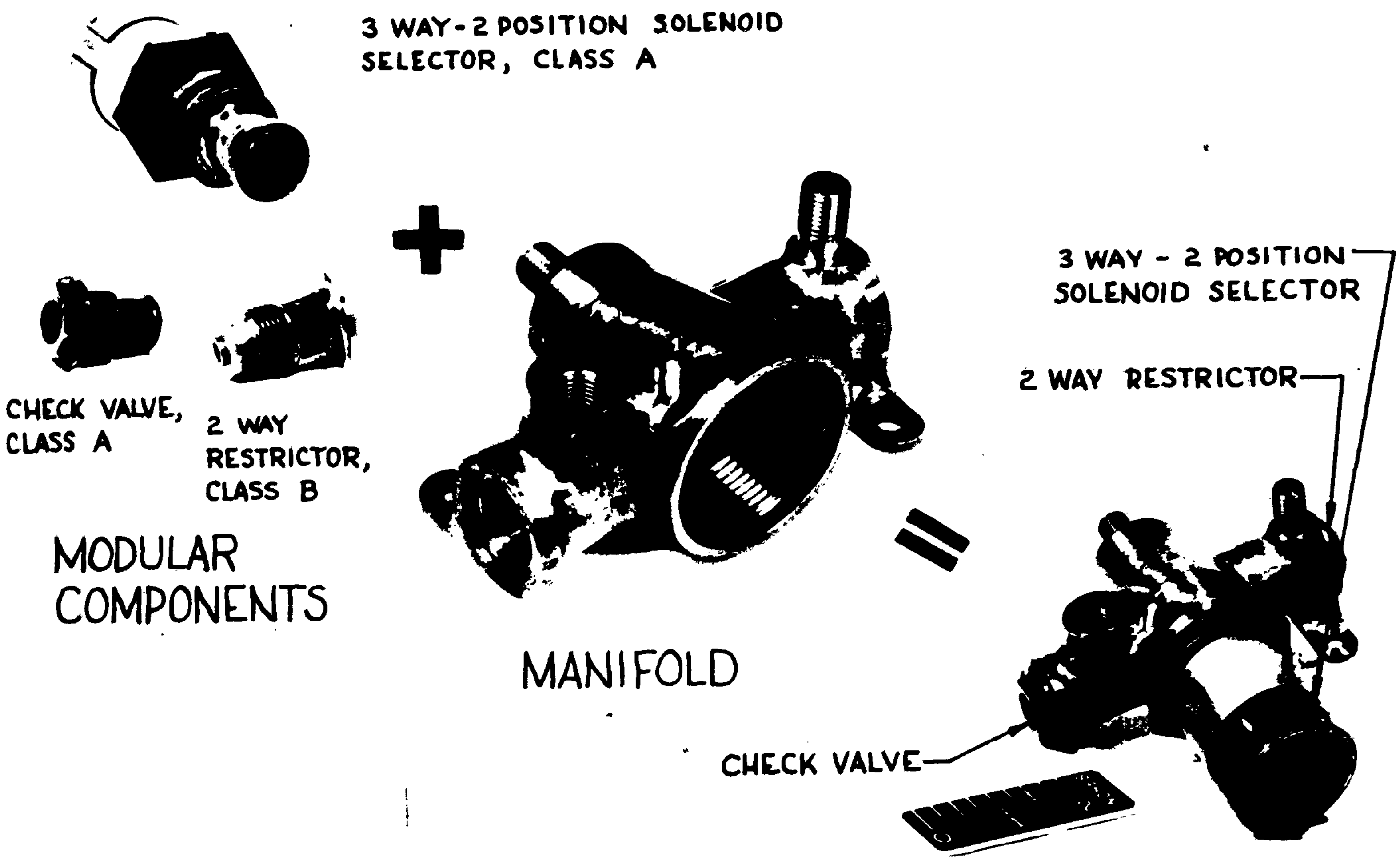


FIGURE III-1
PACKAGING MODULAR HYDRAULIC COMPONENTS

(2) Since modular components are standard and have been qualified, development testing and qualification of new packages are reduced to a minimum.

(3) Since components are developed, the required time from package concept to production is considerably reduced.

(4) Changes to a packaged system still require a penalty; however, by having standard qualified components to use when making changes, the penalty is reduced.

(5) The fabrication of packages in small quantities is expensive, however some cost saving is realized in reduced qualification costs.

(6) Because of initial design flexibility in packaging modular components, location is not unduly penalized. However, since packages are larger than individual components, space must be considered in the initial design when locating a package.

(7) The risk of developing new packages has been considerably reduced by providing standard qualified components.

(8) For the same reason, logistic problems will be substantially reduced.

C. Manifold Design and Packaging Methods

This phase of the program was concerned with developing reliable methods for designing manifolds and packaging of modular components for use in 4,000 psi, 450°F hydraulic systems. To accomplish this the following studies were made:

(1) Representative airplane systems were studied and subsystems that could be packaged were determined. Based on this study, four hydraulic test packages, utilizing representative modular components, were designed and fabricated. The purpose of fabricating the packages was to permit testing of modular components in combination with other modular components as a supplement to individual component qualifications. In addition, package fabrication allowed Chance Vought to gain experience and insight into design requirements and fabrication problems inherent in packaging several modular components.

(2) Techniques of servicing packages were studied and one method demonstrated by package #5.

(3) Methods that make the manifold design easier, lighter and less expensive were studied. This study included such items as the use of permanent plugs, castings and forgings.

(4) Means of locking modular valves in manifold cavities were studied and an acceptable method determined.

(5) Use of thread lubricants was studied to help reduce high component installation torques caused by the high force required to deflect metallic seals.

(6) Based on the above studies, a proposed military specification was prepared and titled "General Specification for the Packaging of Modular Hydraulic Components." This specification is included as Appendix III-1. The specification contains information about designing manifolds, testing manifolds and testing packages, and is one of the important end products of the packaging studies.

Detailed results of the studies are discussed in the following paragraphs.

Package #1 - The first manifold was designed to simulate an arresting gear control package. Figure III-2 shows the schematic of the system and denotes the modular components that are used. CVS-54175 shows the detail design of the manifold and Figures III-3 and III-4 show the finished product. This manifold was designed such that it could be made as a casting or forging so that maximum weight could be saved. However, it was machined from forged titanium (6AL-4V) bar stock because of high costs involved in casting or forging a single unit. Titanium was chosen to demonstrate a minimum weight manifold and to investigate performance of titanium as a manifold material. Both male and female ports were utilized on the manifold design for test purposes.

No problems were encountered in the machining of the titanium in respect to holding close tolerances and the $\sqrt[32]{}$ finishes required for the seal grooves. Boring the larger size cavities for the valves caused some tool springback, but keeping the cutting tools sharp and using a lot of coolant resulted in dimensions within tolerance (i.e., $^{+.003}_{-.000}$ on diameters and 0.002 FIR on concentricities) with a minimum of effort. Profile milling caused some difficulty as evidenced by chatter and tool marks on the outside surface. The part was X-rayed and there were no visible defects.

The manifold successfully withstood proof pressure of 6,000 psi at 450°F, and was then tested in accordance with Chance Vought specification AER-AVO-53720-0-181 (see Appendix III-2). A summary of the #1 package test results is given in Appendix III-6. It will be noted that some troubles were epxerienced with the selector valves and restrictor; however, later valve improvements eliminated these problems. Slight wear marks were noted on the cavity of Hi-Ceal seats but not enough to cause leakage. No problems were experienced with either installing or removing the components. From Chance Vought's experience with this package, the design was very satisfactory, and the 6Al-4V titanium material performed well. However, test results were not conclusive in establishing the wear properties of titanium as a Hi-Ceal seat material because the package was not subjected to pressure impulse tests. It is the opinion of Chance Vought (see discussion of titanium in Part IV - Materials) that cavity seat wear during pressure impulse

#1 TEST PACKAGE

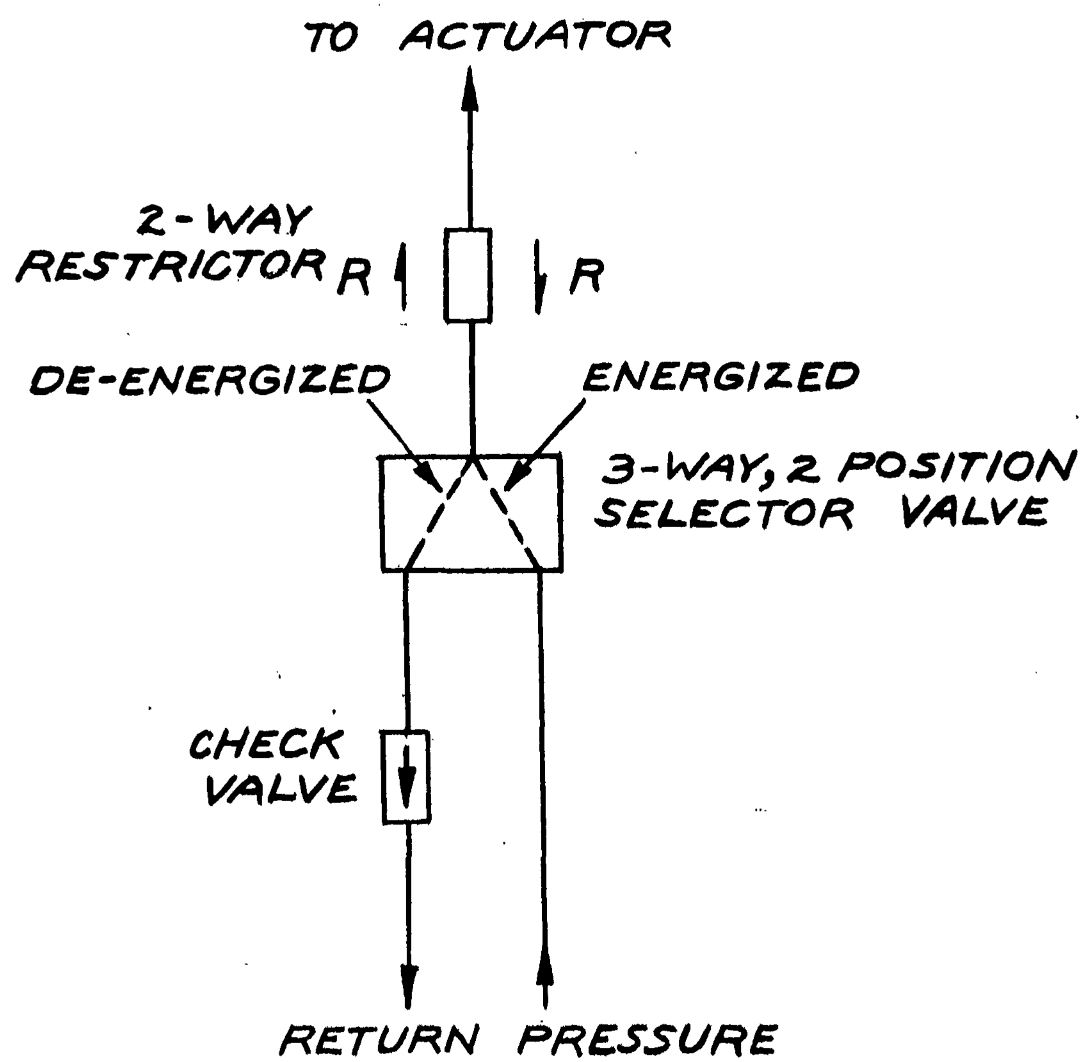
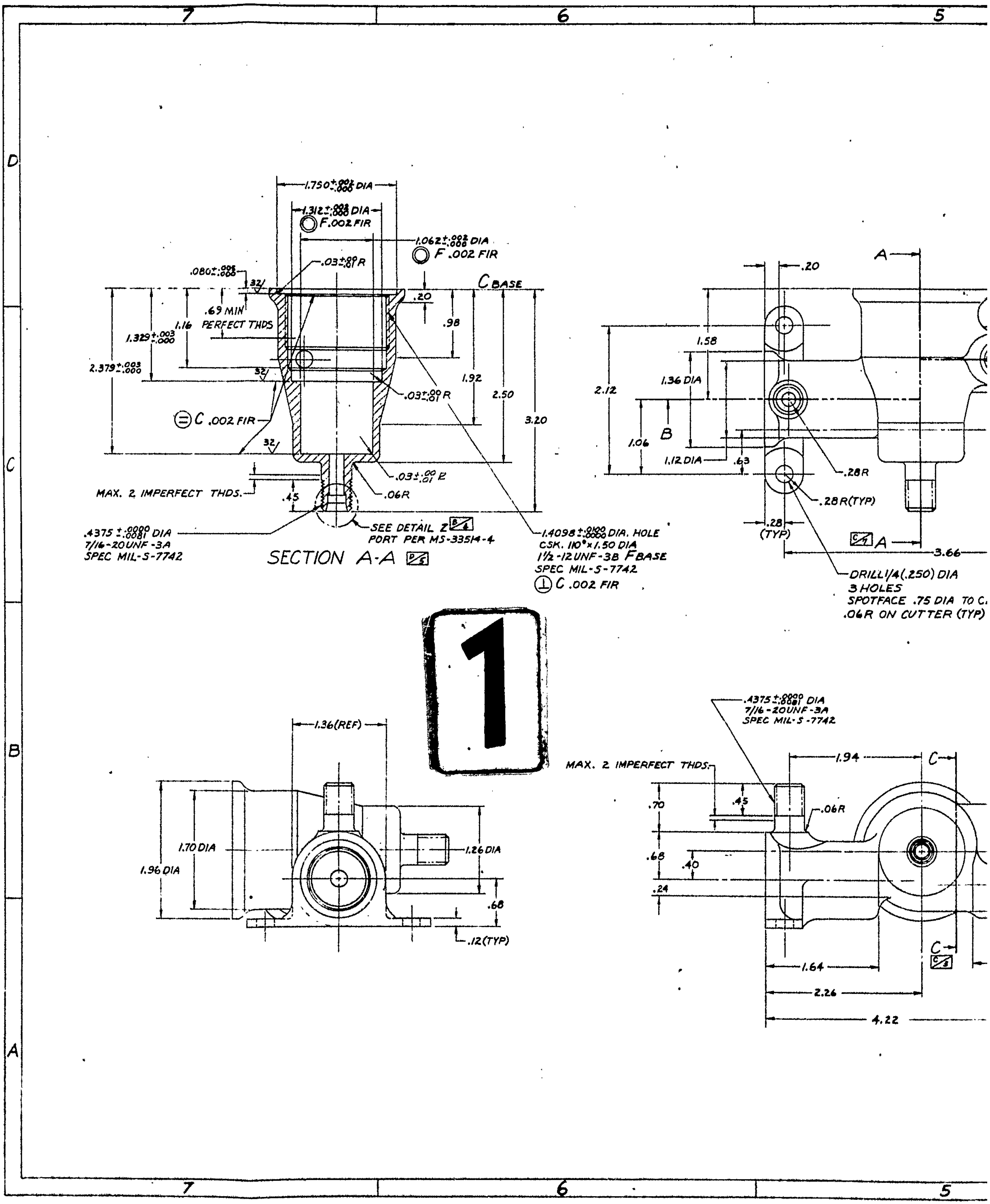
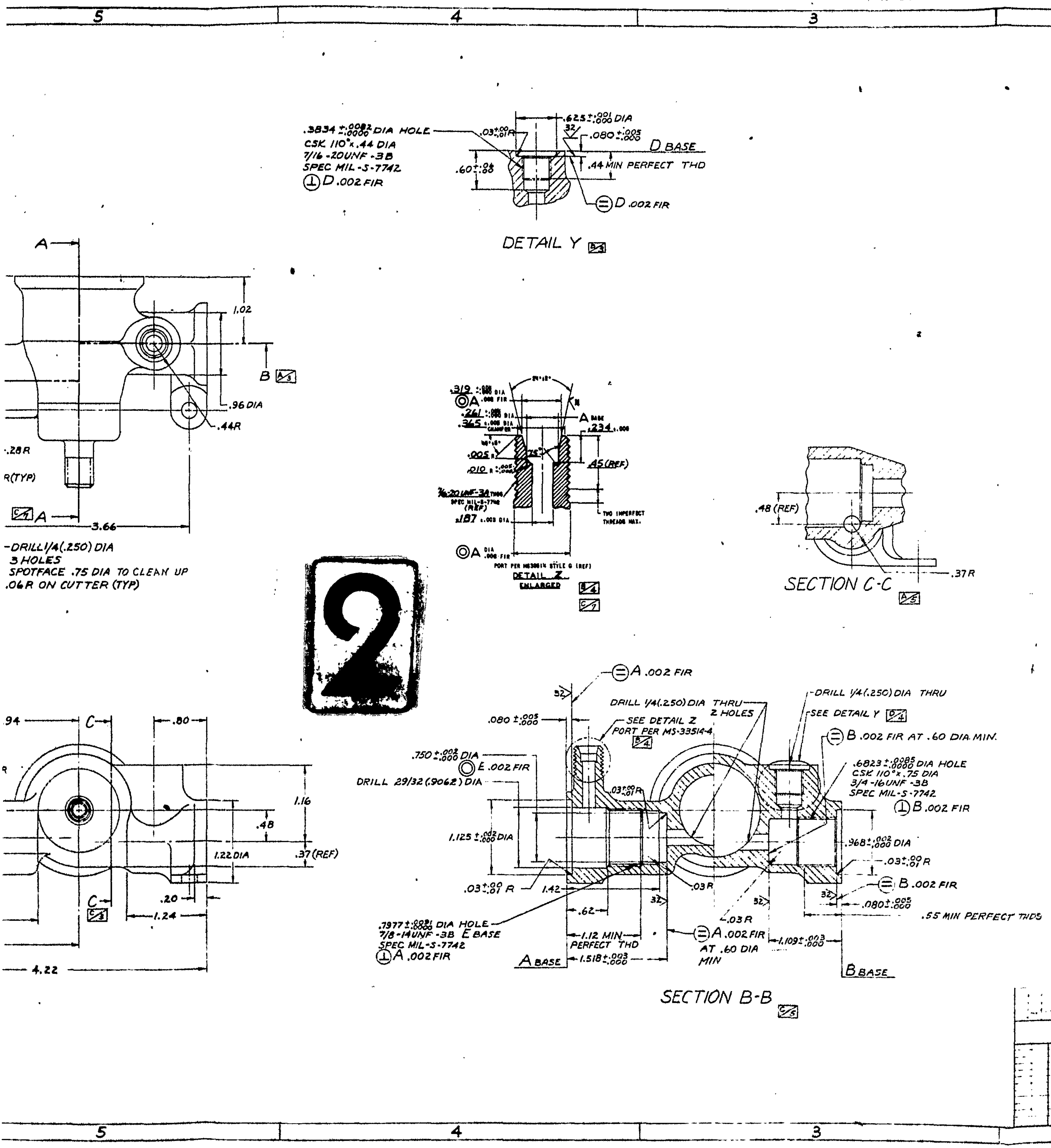


FIGURE III-2

CVS-54125

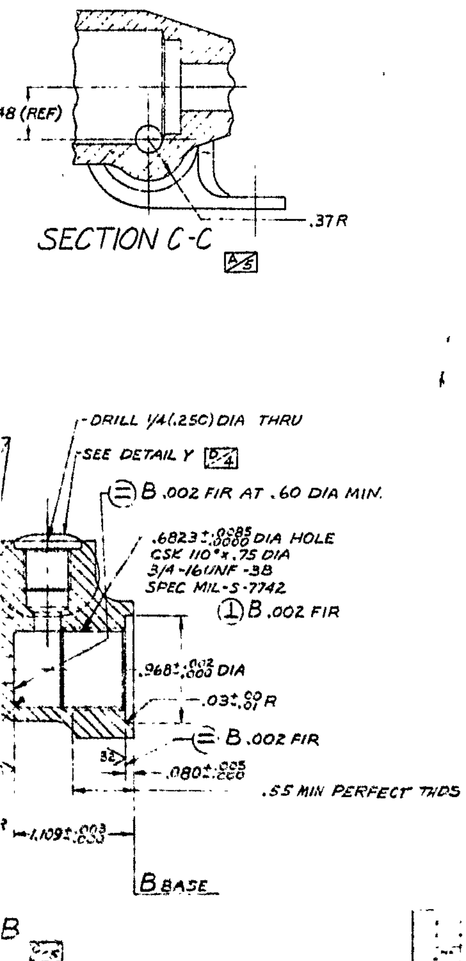

SECTION A-A
1.750 DIA
1.312 DIA
F .002 FIR
1.062 DIA
C BASE
.69 MIN PERFECT THDS
C .002 FIR
MAX. 2 IMPERFECT THDS.
.4375 DIA
7/16-20UNF-3A
SPEC MIL-S-7742
SEE DETAIL Z
PORT PER MS-33514-4
1.4098 DIA. HOLE
CSK. 110° x 1.50 DIA
1½-12UNF-3B F BASE
SPEC MIL-S-7742
C .002 FIR
DRILL 1/4 (.250) DIA
3 HOLES
SPOTFACE .75 DIA TO C
.06R ON CUTTER (TYP)
1.36 (REF)
1.70 DIA
1.96 DIA
1.26 DIA
.12 (TYP)
1.36 DIA
1.12 DIA
.28R (TYP)
1.94
1.64
2.26
4.22
3.66
1



5
4
3
.3834 +.0002 -.0000 DIA HOLE
CSK 110° x .44 DIA
7/16 -20UNF -3B
SPEC MIL-S-7742
D .002 FIR
.625 +.001 -.000 DIA
.080 +.005 -.000
D BASE
.44 MIN PERFECT THD
D .002 FIR
DETAIL Y
1.02
B
.96 DIA
.44R
.28R
R(TYP)
A
3.66
DRILL 1/4(.250) DIA
3 HOLES
SPOTFACE .75 DIA TO CLEAN UP
.06R ON CUTTER (TYP)
.45(REF)
TWO IMPERFECT THREADS MAX.
PORT PER MS33514 STYLE G (REF)
DETAIL Z
ENLARGED
.48 (REF)
.37R
SECTION C-C
2
A .002 FIR
DRILL 1/4(.250) DIA THRU
2 HOLES
DRILL 1/4(.250) DIA THRU
SEE DETAIL Y
SEE DETAIL Z
PORT PER MS-33514-4
B .002 FIR AT .60 DIA MIN.
.080 +.005 -.000
.750 +.002 -.000 DIA
E .002 FIR
DRILL 29/32 (.9062) DIA
.6823 +.0085 -.0000 DIA HOLE
CSK 110° x .75 DIA
3/4 -16UNF -3B
SPEC MIL-S-7742
B .002 FIR
1.125 +.002 -.000 DIA
.968 +.002 -.000 DIA
.03 +.00 -.01 R
B .002 FIR
.080 +.005 -.000
.55 MIN PERFECT THDS
1.42
.62
.03R
.7977 +.0021 -.0000 DIA HOLE
7/8 -14UNF -3B E BASE
SPEC MIL-S-7742
A .002 FIR
1.12 MIN PERFECT THD
A .002 FIR
AT .60 DIA
MIN
1.109 +.003 -.000
A BASE
1.518 +.003 -.000
B BASE
SECTION B-B
.94
C
.80
1.16
.48
1.22 DIA
.37(REF)
.20
1.24
4.22

NOTES

1. BREAK ALL SHARP EDGES
2. MACHINE FINISH 125 EXCEPT AS NOTED
3. UNMARKED FILLET RADII .25R
   CORNER RADII .12R
4. SYMBOLS ◎ FOR CONCENTRICITY, ⊥ FOR SQUARENESS AND ⊜ FOR PARALLELISM SHOW REQUIREMENTS WITHIN LIMITS SPECIFIED. SURFACES AFFECTED ARE IDENTIFIED BY LEADERS OR SAME LETTER SYMBOLS



3

CVS-54175-1	MANIFOLD	2 1/2 x 3 1/4 S NO. FORGED BILLET	6AL-4V Ti MIL-T-9047C CLASS 5	130,000	NONE	0.78	

±.01 ~ ±30'

34x77

2x SIZE & NOTED

ROSEHILL 53732

VOUGHT AIRCRAFT INCORPORATED DALLAS TEXAS

MANIFOLD
ARRESTING GEAR CIRCUIT
MODULAR HYDRAULIC

CVS-54175

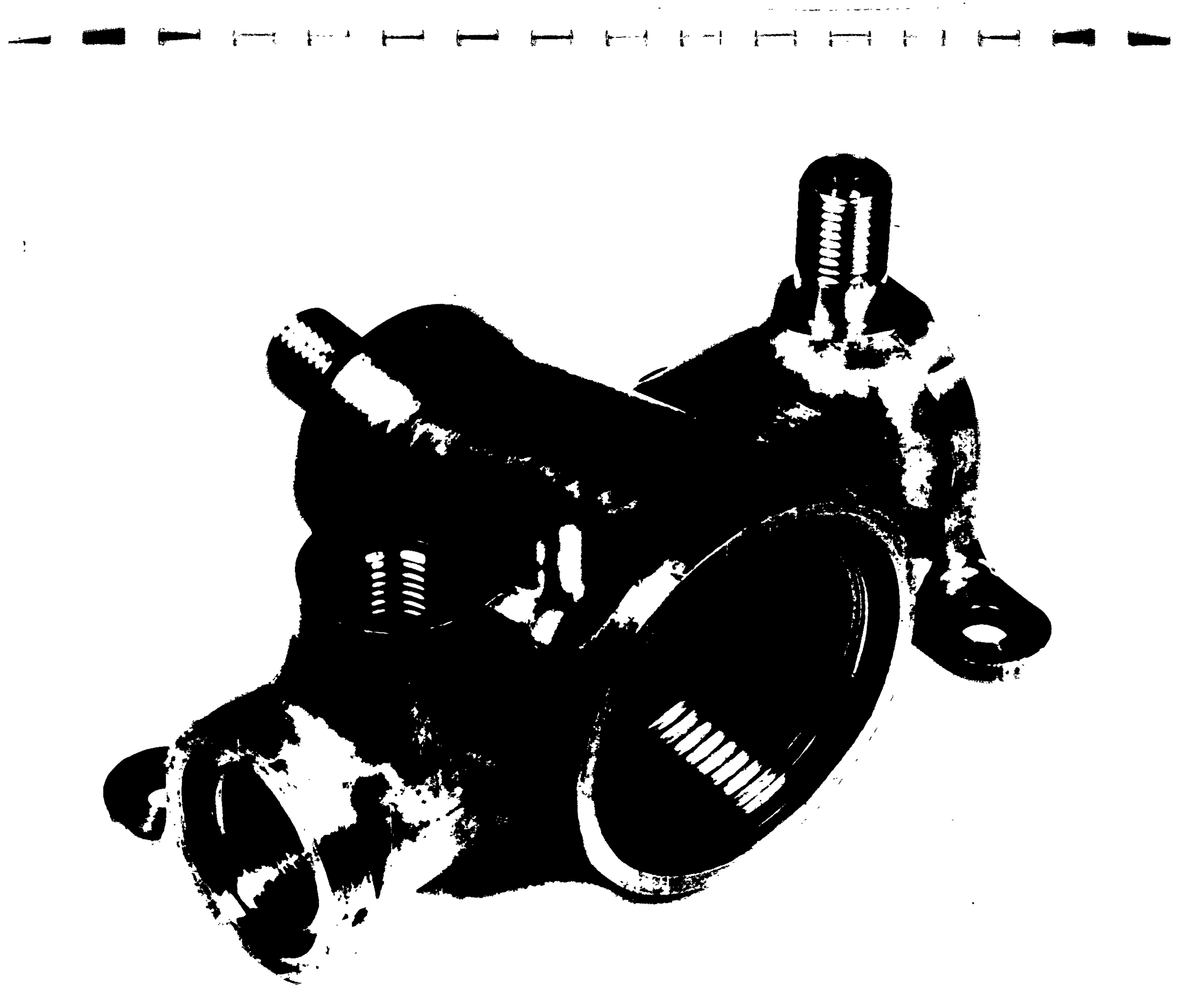


FIGURE III-3
#1 PACKAGE MANIFOLD

FIGURE III-4
#1 PACKAGE

cycling will be much more severe with titanium than with stainless steel. To establish an endurance limit of titanium alloys further testing is required.

A considerable weight saving was achieved using titanium in lieu of stainless steel. Wall thicknesses for both materials would have to be 0.10 minimum, due to fabrication considerations. As a result, the titanium manifold weighed 0.68 pounds in contrast to 1.20 pounds if it had been made of stainless steel.

Package #2 - Package #2 is a packaged actuator design which utilizes modular valves that are often associated with actuator operation. A schematic of the package is shown in Figure III-5. Detail drawings of the end cap and the actuator assembly are shown on drawings CVS-54545 and CVS-54542, respectively. Pictures of the finished product are shown in Figures III-6, III-7 and III-8.

Four basic configurations evolved from preliminary layout work done on this package. These configurations are shown and briefly discussed in Figures III-9, III-10, III-11 and III-12. It was considered advantageous to design a manifolded end cap to be removable from the cylinder barrel, since this reduces cost of the manifold and facilitates testing.

All the manifold configurations except Figure III-12 indicated that the cylinder bore diameter should not be less than approximately 2 inches. This limitation was caused by the physical interference of the valves and by the off-set drilling required for internal porting. Most of the manifold designs required at least one external plug to plug connecting passages between valves. The "Lee plug" was selected for this purpose and performed satisfactorily. It is discussed in later paragraphs. It can be seen that valves located to give good accessibility in an actual airframe installation usually result in a little heavier installation. This is because the components cannot be oriented to take advantage of maximum weight removal.

All major parts of the actuator are of 17-4PH stainless steel. The original design concept showed a chromium-plated piston and rod sliding on an electro-less nickel-plated barrel. The nickel plate was to have provided high wear resistance and reasonable frictional qualities. However, further investigation revealed that chromium rubbing on electroless nickel plate without the benefit of a good lubricant resulted in a coefficient of friction of 0.43. Therefore, it was decided to use a chromium-plated piston and rod in combination with bare 17-4PH stainless steel for the barrel. This gives a coefficient of friction of approximately 0.19 to 0.23 but will reduce the life of the actuator considerably due to the relative softness of the stainless steel in comparison with nickel plate. As a comparison, 17-4PH has a hardness of Rc 40/46, electroless nickel plate Rc 70, and hard anodized aluminum Rc 50/58.

Samples of improved Viton O-rings were procured for use in package #2 actuator from Parker Seal Company, Linear, Inc., and Precision Rubber Products. All three seal manufacturers indicated that they felt these O-rings would be satisfactory. Chance Vought's experience with these viton piston and rod O-rings was that they performed very satisfactorily from room temperature to 450°F. While operating at 450°F, they would take a permanent

#2 TEST PACKAGE

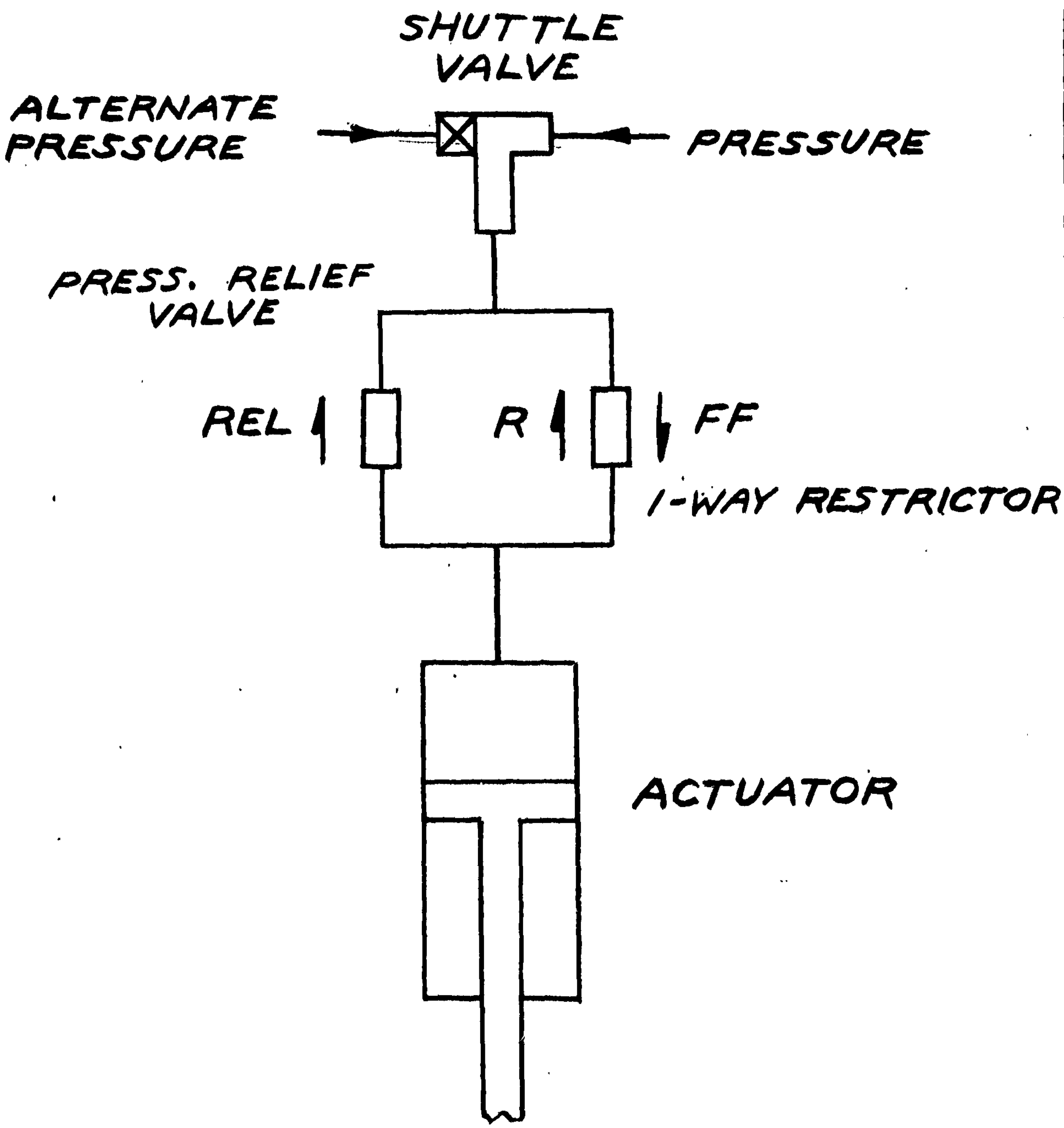
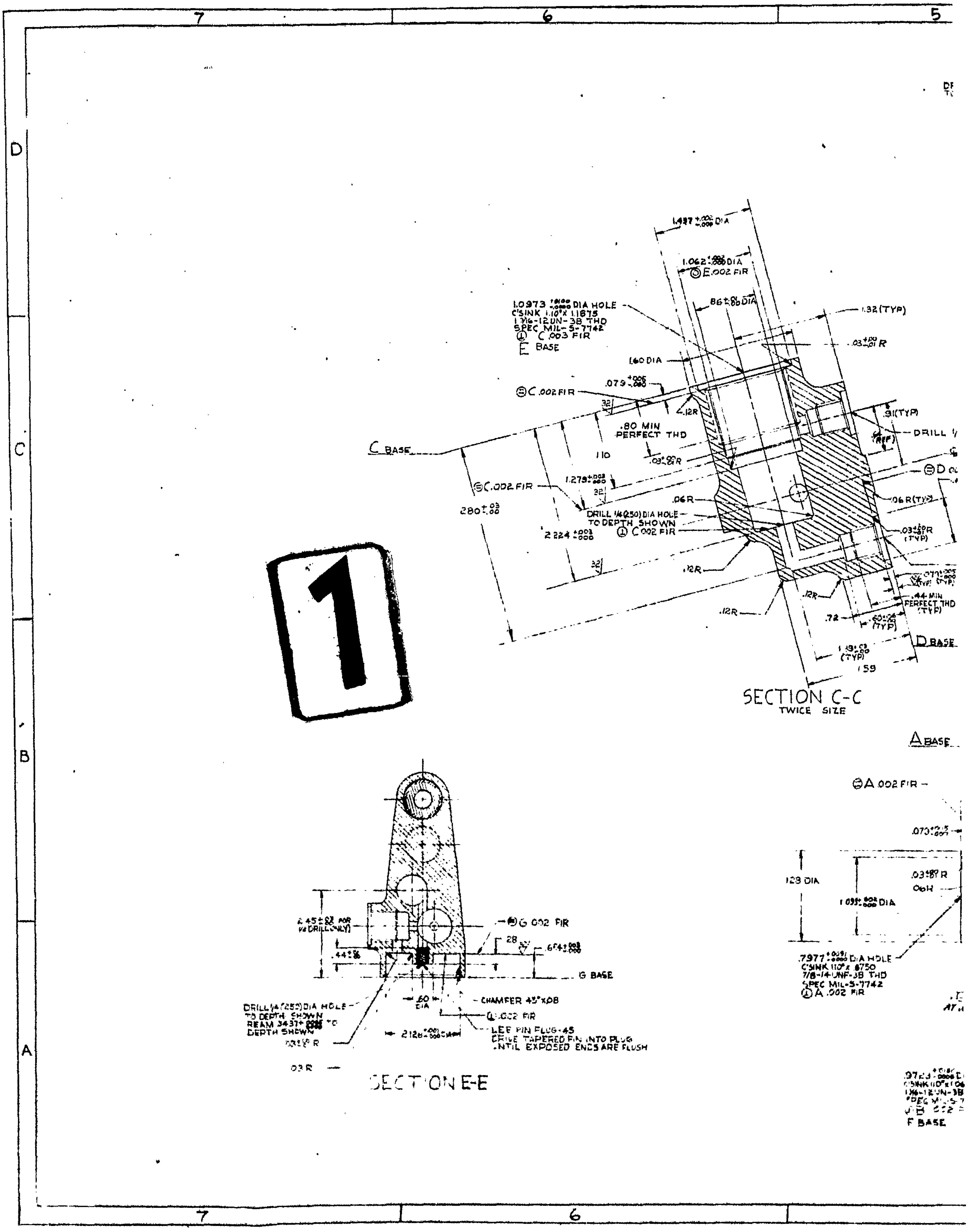
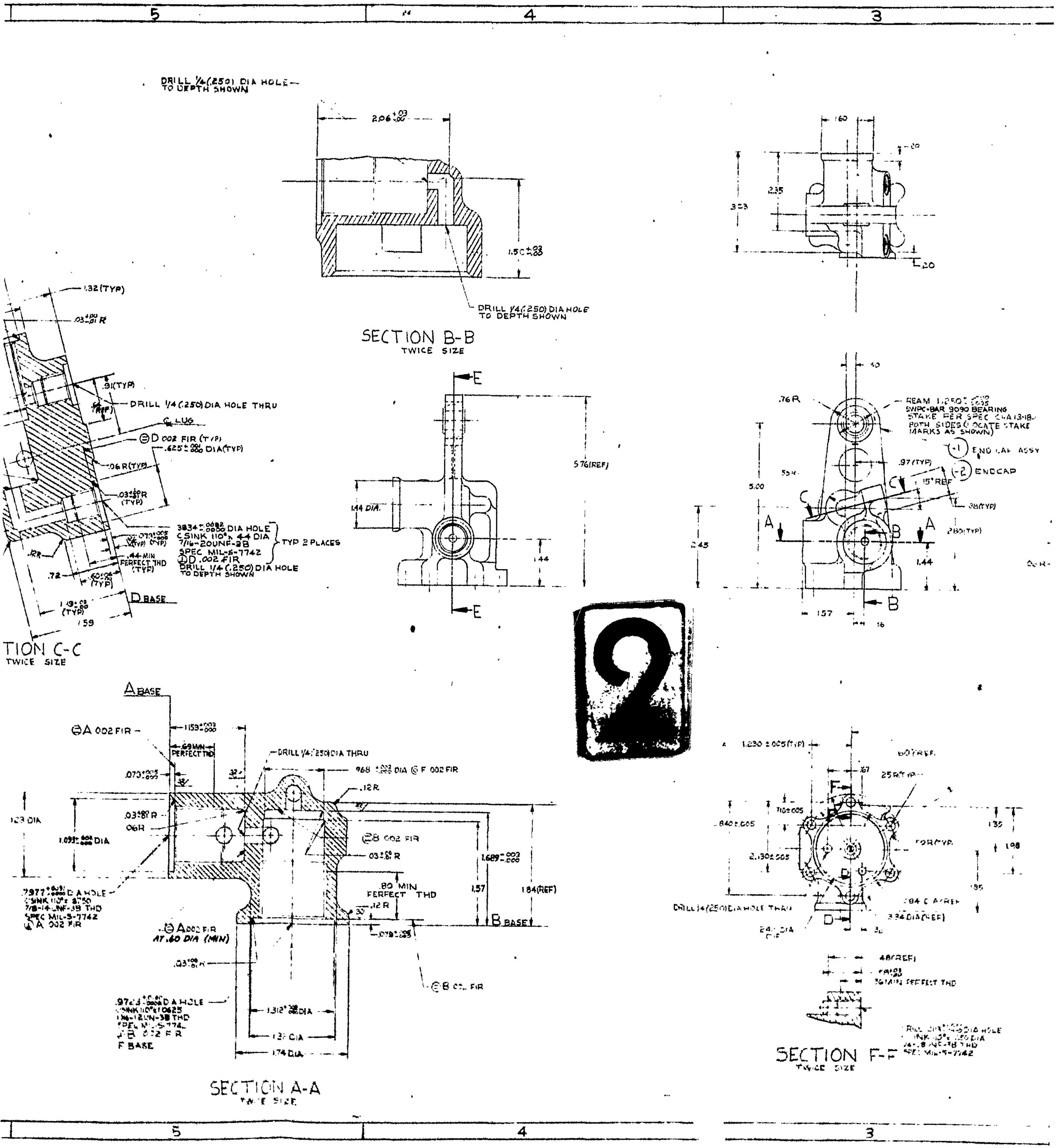


FIGURE III-5

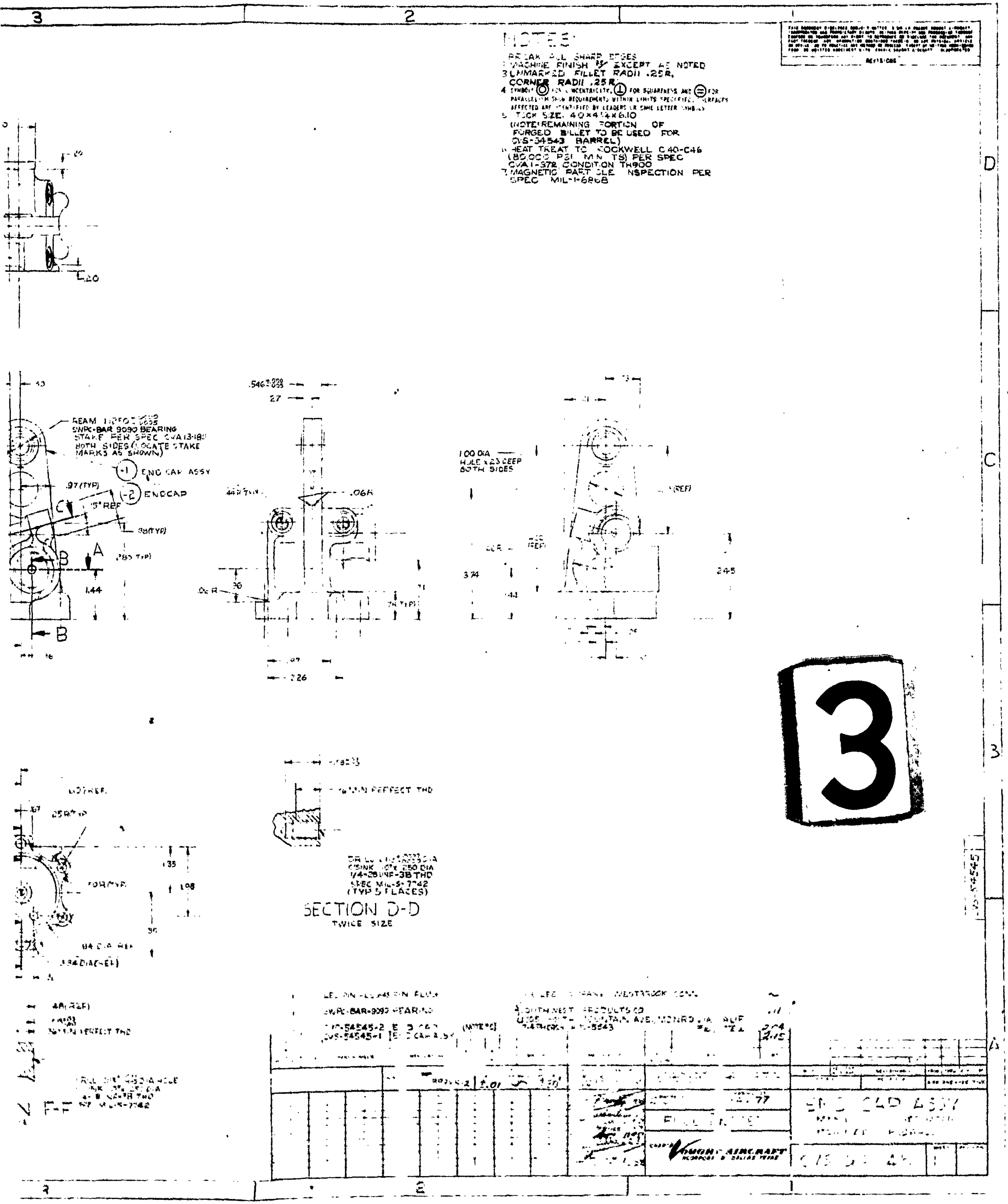


SECTION C-C
TWICE SIZE

SECTION E-E



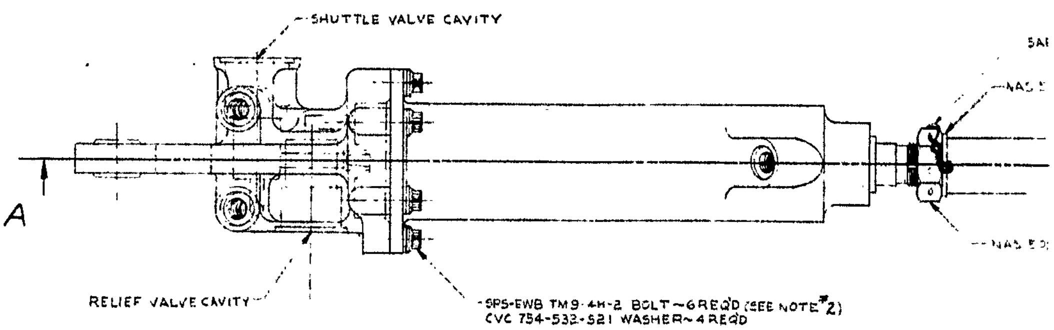
5
4
3
DRILL 1/4(.250) DIA HOLE TO DEPTH SHOWN
SECTION B-B
TWICE SIZE
DRILL 1/4(.250) DIA HOLE THRU
DRILL 1/4 (.250) DIA HOLE TO DEPTH SHOWN
TYP 2 PLACES
D BASE
TION C-C
TWICE SIZE
5.76(REF)
REAM 1.2507
SWPC-BAR 9090 BEARING
END CAP ASSY
ENDCAP
A BASE
DRILL 1/4(.250)DIA THRU
PERFECT THD
.80 MIN PERFECT THD
B BASE
F BASE
SECTION A-A
TWICE SIZE
SECTION F-F
TWICE SIZE
5
4
3


NOTES:
BREAK ALL SHARP EDGES
MACHINE FINISH EXCEPT AS NOTED
3 UNMARKED FILLET RADII .25R, CORNER RADII .25R
4 SYMBOLS FOR CONCENTRICITY, FOR SQUARENESS AND FOR PARALLELISM SHOW REQUIREMENTS WITHIN LIMITS SPECIFIED. SURFACES AFFECTED ARE IDENTIFIED BY LEADERS OR SAME LETTER SYMBOLS
5 STOCK SIZE. 4.0×4¼×6.10
(NOTE: REMAINING PORTION OF FORGED BILLET TO BE USED FOR CVS-54543 BARREL)
6 HEAT TREAT TO ROCKWELL C 40-C46 (180,000 PSI MIN TS) PER SPEC CVA1-372 CONDITION TH900
7 MAGNETIC PARTICLE INSPECTION PER SPEC MIL-I-6868
REAM
SWPC-BAR 9090 BEARING
STAKE PER SPEC CVA13-18
BOTH SIDES (LOCATE STAKE MARKS AS SHOWN)
-1 END CAP ASSY
-2 ENDCAP
1.00 DIA HOLE x.23 DEEP BOTH SIDES
SECTION D-D
TWICE SIZE
CVS-54545-2 END CAP
CVS-54545-1 END CAP ASSY
SOUTHWEST PRODUCTS CO
END CAP ASSY
VOUGHT AIRCRAFT

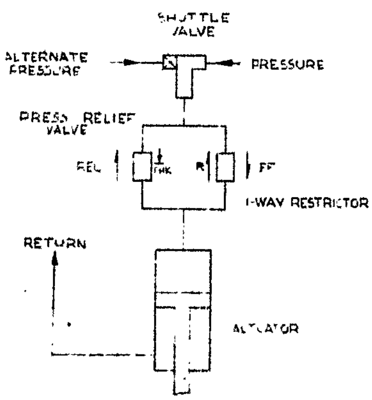
3

5 4

D



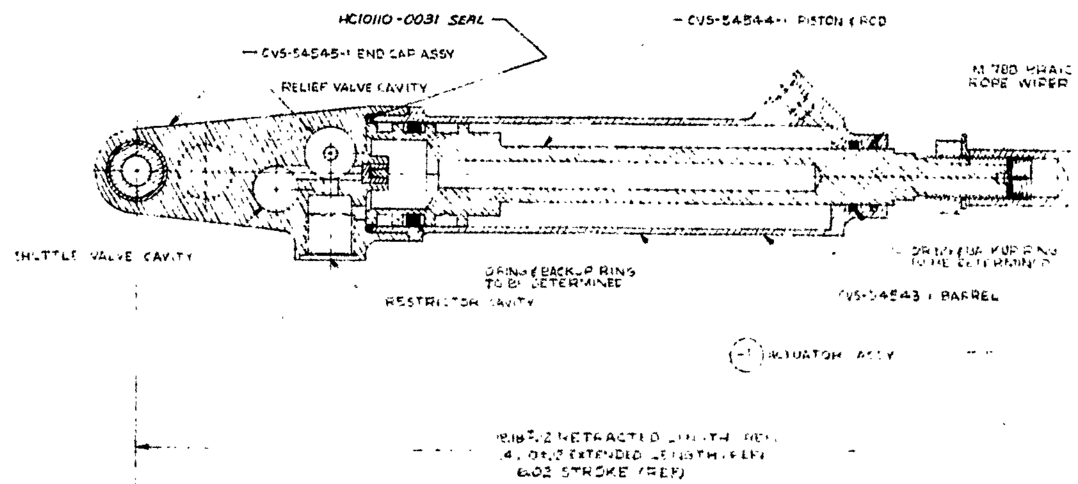
C



1

B

SCHEMATIC

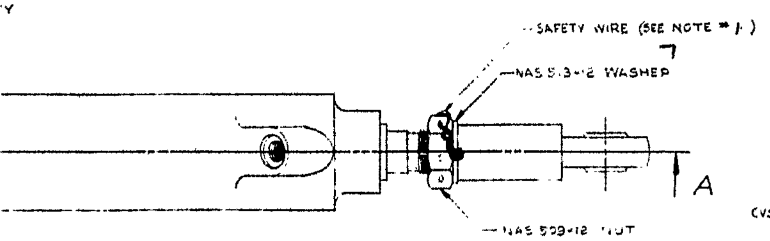


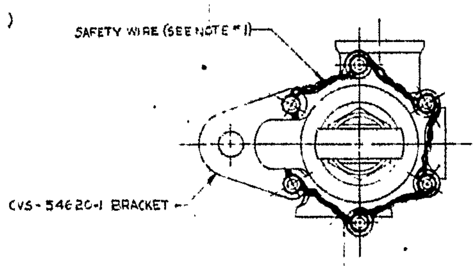
A

A-A

5 4

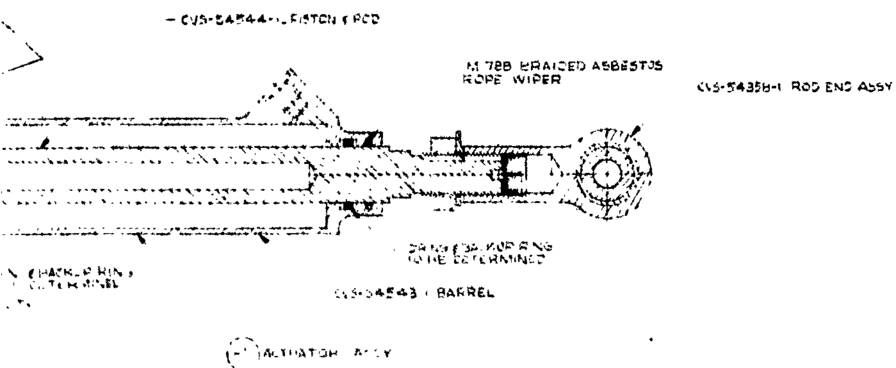
NOTES:
1. SAFETY WIRE PER SPEC CSA 13-180
2. TORQUE BOLTS 120-135 IN LBS
3. OPERATING PRESSURE 4000 PSI
4. THIS UNIT TO OPERATE WITH MLO-8200
5. OPERATING TEMP. -65°F TO +450°F FLUID
   -65°F TO +650°F AMBIENT
6. A. CYLINDER TO BE ASSEMBLED UNDER THE DIRECTION OF THE HYDRAULIC TEST LAB
   B. ALL PARTS INCLUDING PACKINGS ARE TO BE THOROUGHLY WASHED WITH MLO-8200 PRIOR TO ASSEMBLY. EXCESSIVE AMOUNTS OF CONTAMINENTS CAN BE FLUSHED FROM ALL PARTS EXCEPT PACKINGS USING STODDARD SOLVENT, SPEC P-S-661 AND PARTS DRIP DRAINED FOLLOWED BY FLUSHING WITH MLO-8200 AND DRIP DRAINED.
   C. AFTER ASSEMBLING, UNIT IS TO BE FLUSHED WITH CLEAN MLO-8200 AND DRIP DRAINED.
   D. UNIT TO BE STORED WITH ALL OPENINGS PROTECTED FROM ENTRY OF CONTAMINATION
   E. UNIT WILL BE ACCEPTANCE TESTED BY THE HYDRAULIC TEST LAB





SPS-EWB TM9-4H-2 BOLT ~ 6 REQD (SEE NOTE #2)
CVC 754-532-521 WASHER ~ 4 REQD

2



A-A

1	HC10110-0031 SEAL (HI-CEAL)	HIGH PRESSURE ENGINEERS, INC., BELTSVILLE, MD	~
1	M 788 BRAIDED ASBESTOS ROPE WIPER 9 x 354	JOHN MANVILLE CORP. NEW YORK N.Y.	~
4	CVC 754-532-521 WASHER		.01
1	NAS 513-12 WASHER		~
1	NAS 509-12 NUT		.09
6	SPS EWB TM9-4H-2 BOLT	STANDARD PRESSED STEEL CO. JENKINTOWN PENN	.08
1	CVS-54620-1 BRACKET		[illegible]
1	CVS-54545-1 END CAP ASSY		2.[illegible]
1	CVS-54544-1 PISTON & ROD		[illegible]
1	CVS-54543-1 BARREL		[illegible]30
1	CVS-54358-1 ROD END ASSY		[illegible]
	[illegible]-54542- ACTUATOR ASSY		7.60

99948-2

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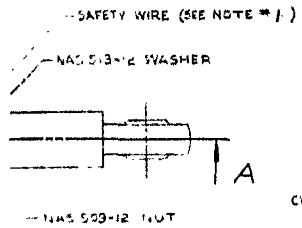
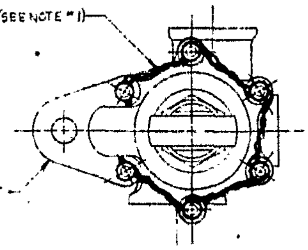
71

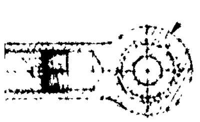
NOTES:

1 SAFETY WIRE PER SPEC C%A 13-180
2 TORQUE BOLTS 120-135 IN LBS
3 OPERATING PRESSURE 4000 PSI
4 THIS UNIT TO OPERATE WITH MLO-8200
5 OPERATING TEMP -65°F TO +450°F FLUID
-65°F TO +650°F AMBIENT
6 A. CYLINDER TO BE ASSEMBLED UNDER THE DIRECTION OF THE HYDRAULIC TEST LAB
B. ALL PARTS INCLUDING PACKINGS ARE TO BE THOROUGHLY WASHED WITH MLO-8200 PRIOR TO ASSEMBLY. EXCESSIVE AMOUNTS OF CONTAMINENTS CAN BE FLUSHED FROM ALL PARTS EXCEPT PACKINGS USING STODDARD SOLVENT, SPEC P-S-661 AND PARTS DRIP DRAINED FOLLOWED BY FLUSHING WITH MLO-8200 AND DRIP DRAINED.
C. AFTER ASSEMBLING, UNIT IS TO BE FLUSHED WITH CLEAN MLO-8200 AND DRIP DRAINED.
D. UNIT TO BE STORED WITH ALL OPENINGS PROTECTED FROM ENTRY OF CONTAMINATION
E. UNIT WILL BE ACCEPTANCE TESTED BY THE HYDRAULIC TEST LAB





3



1	HC10110-0031 SEAL (HI-CEAL)	HIGH PRESSURE ENGINEERS, INC., BELTSVILLE, MD	~
1	BRAIDED ASBESTOS ROPE WIPER 18 x 354	JOHN MANVILLE CORP, NEW YORK N.Y	~
4	NC 754-532-521 WASHER		.01
1	NAS 513-12 WASHER		~
1	NAS 509-12 NUT		.09
6	SP BOLT	STANDARD PRESSED STEEL CO. JENKINTOWN PENN	.08
1	CVS-54620-1 BRACKET		.10
1	CVS-54545-1 END CAP ASSY		2.5
1	CVS-54544- PISTON & ROD		.037
1	CVS-54543- BARREL		2.30
1	54358- ROD END ASSY		.50
	CVS-54542- ACTUATOR ASSY		7.60

53995-2

ACTUATOR ASSY
MANIFOLDED
MODULAR HYDRAULIC

VOUGHT AIRCRAFT
INCORPORATED DALLAS TEXAS

CVS-54542

1 of 1

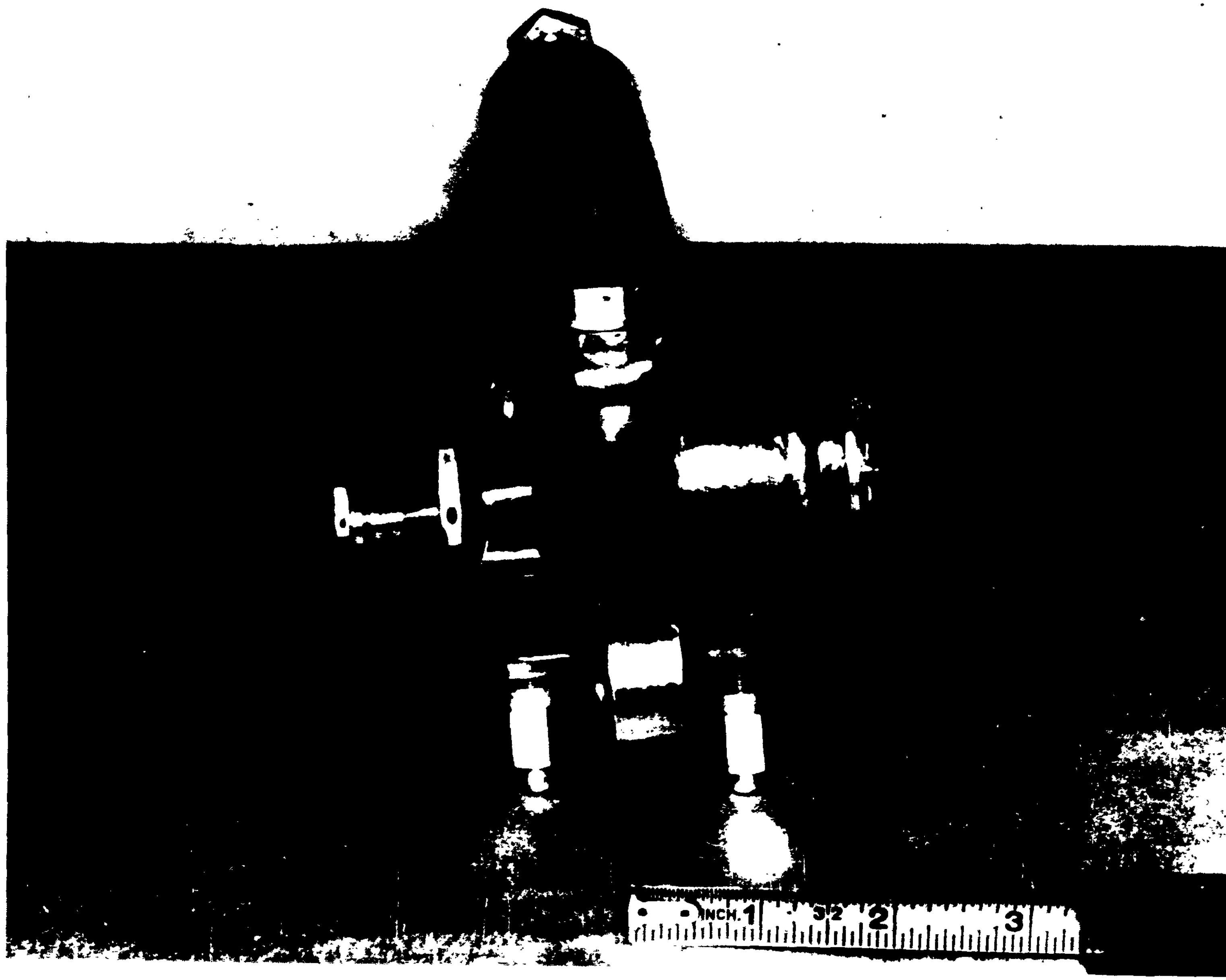


FIGURE III-6
#2 PACKAGE

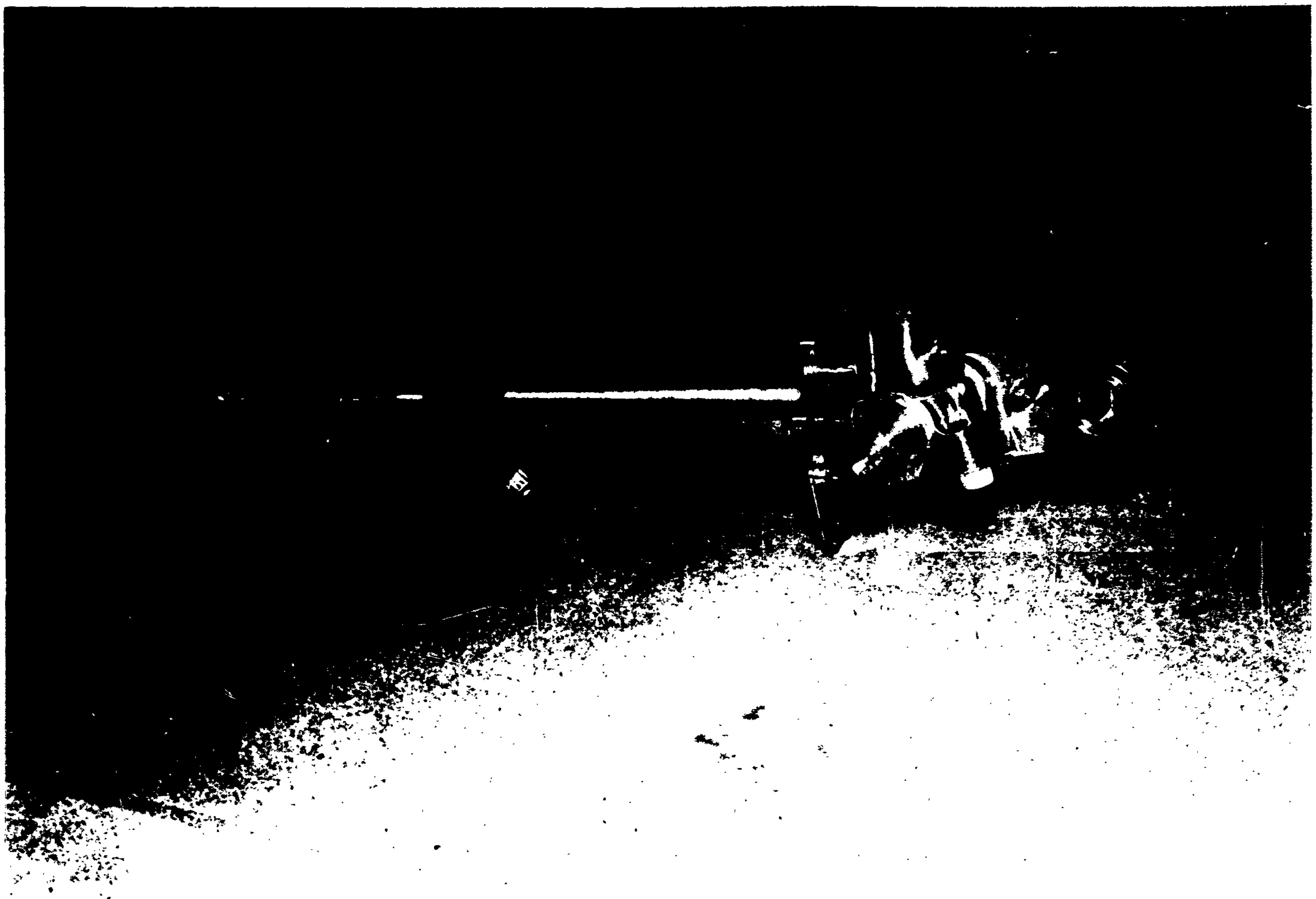


FIGURE III-7

CVS-54542-1 MODULAR HYDRAULIC ACTUATOR WITH END CAP MANIFOLD ASSY. PACKAGE #2, 4 GPM, 3 MODULES (PRESSURE RELIEF VALVE, ONE-WAY RESTRICTOR, & SHUTTLE VALVE).

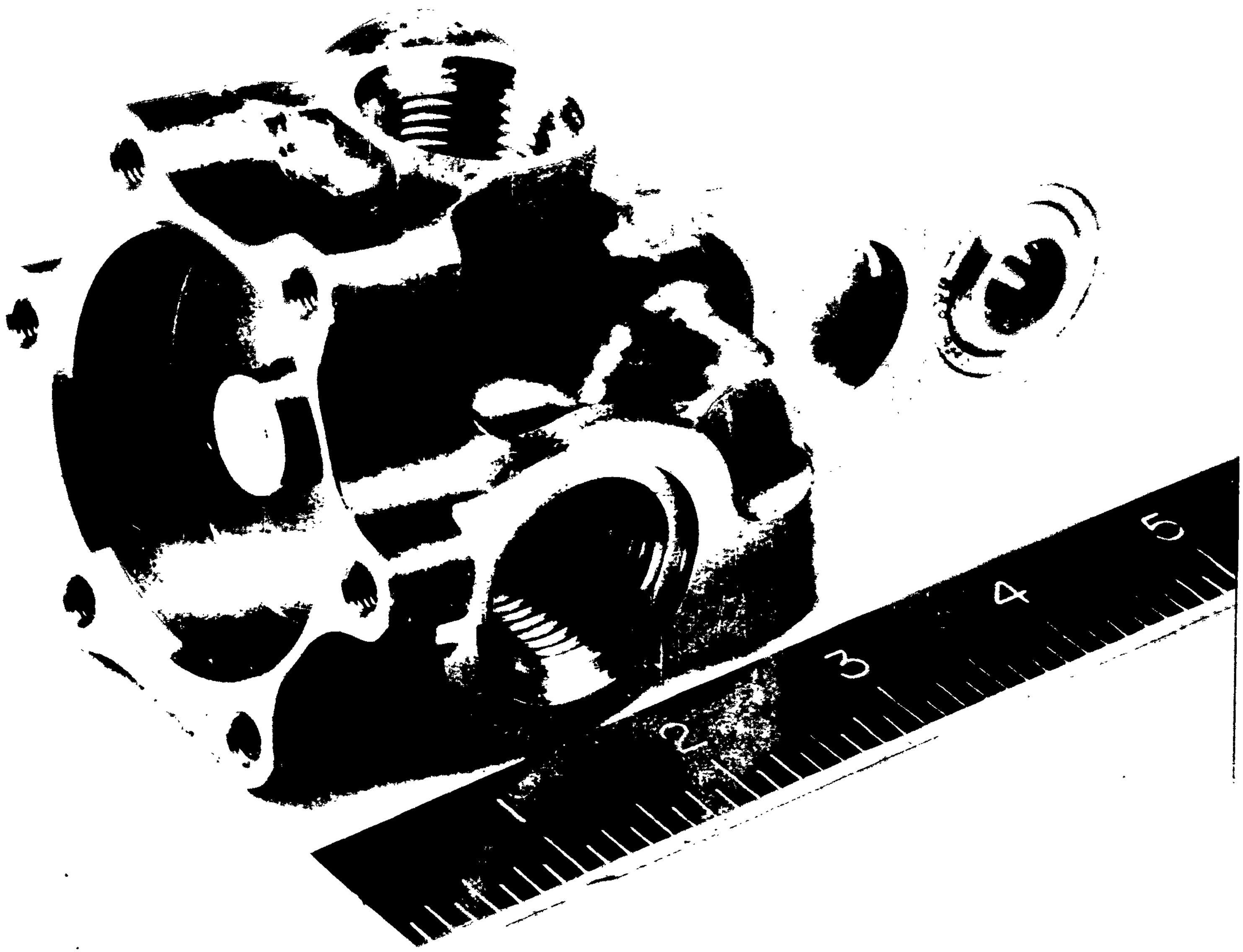


FIGURE III-8

CVS-54545 MODULAR HYDRAULIC END CAP ASSY FOR USE WITH THREE ¼ GPM MODULAR COMPONENTS (PRESSURE RELIEF VALVE, ONE-WAY RESTRICTOR, & SHUTTLE VALVE).

FIGURE III-9

CONFIGURATION "A"

Valve installation between actuator attach lug and barrel -- this configuration results in a minimum space requirement in the lug area but will lengthen the actuators considerably. Since the valves are installed on three faces of the cap, accessibility would be limited in an actual installation. Minimum weight is achieved since a large percentage of material around the valves can be removed. As in most of the manifolded designs, the configuration lends itself toward casting or forging. Since the actuator loads flow around the valve cavities, stress concentrations and/or sealing problems may be encountered.

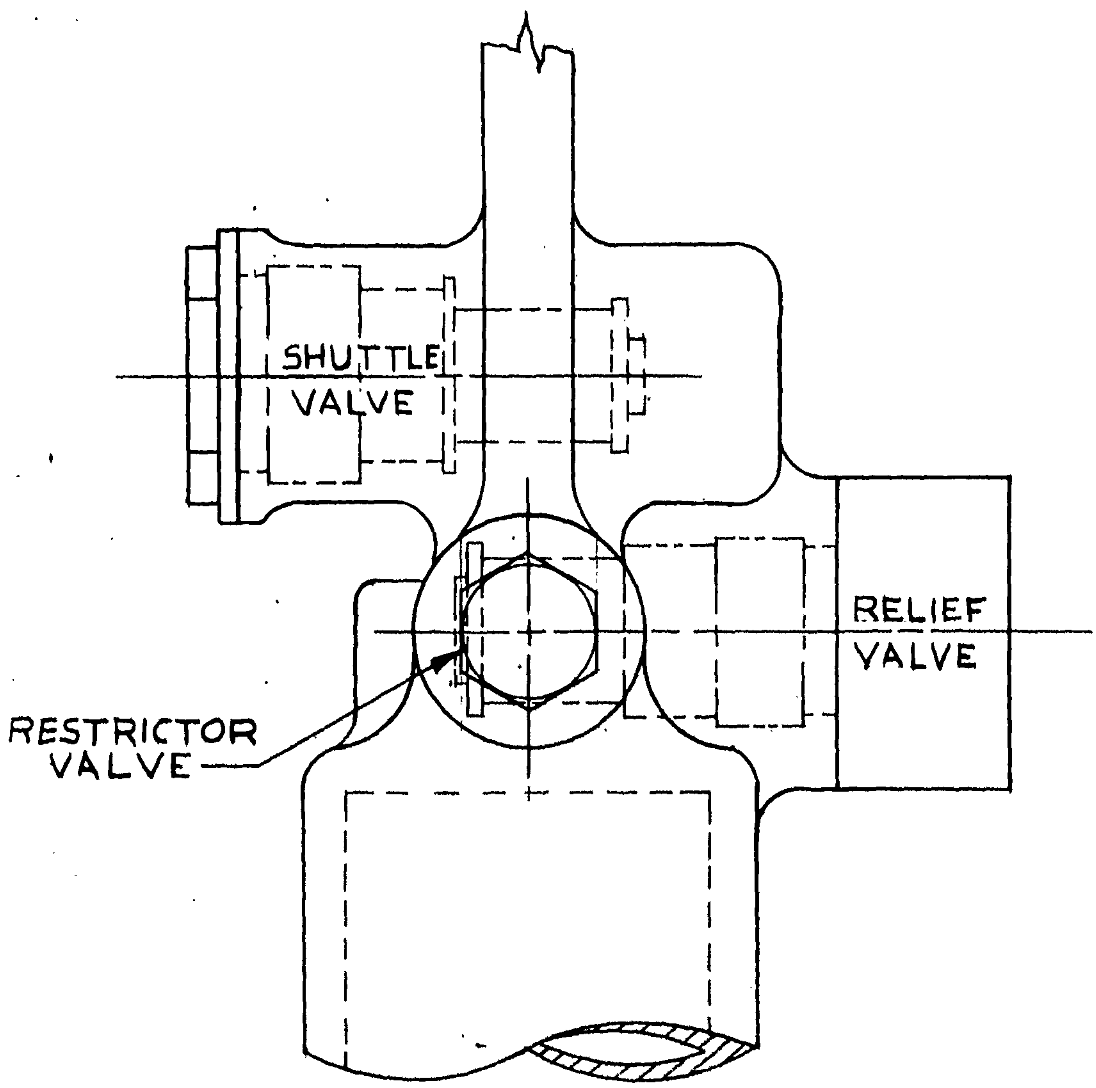


FIGURE III-10

CONFIGURATION "B"

Valve installation in lug area, i.e., between and along side of lugs -- this configuration is similar to "A" in respect to space, length of actuator and weight. Maintenance would be poor since the actuator would have to be disconnected to remove one of the valves.

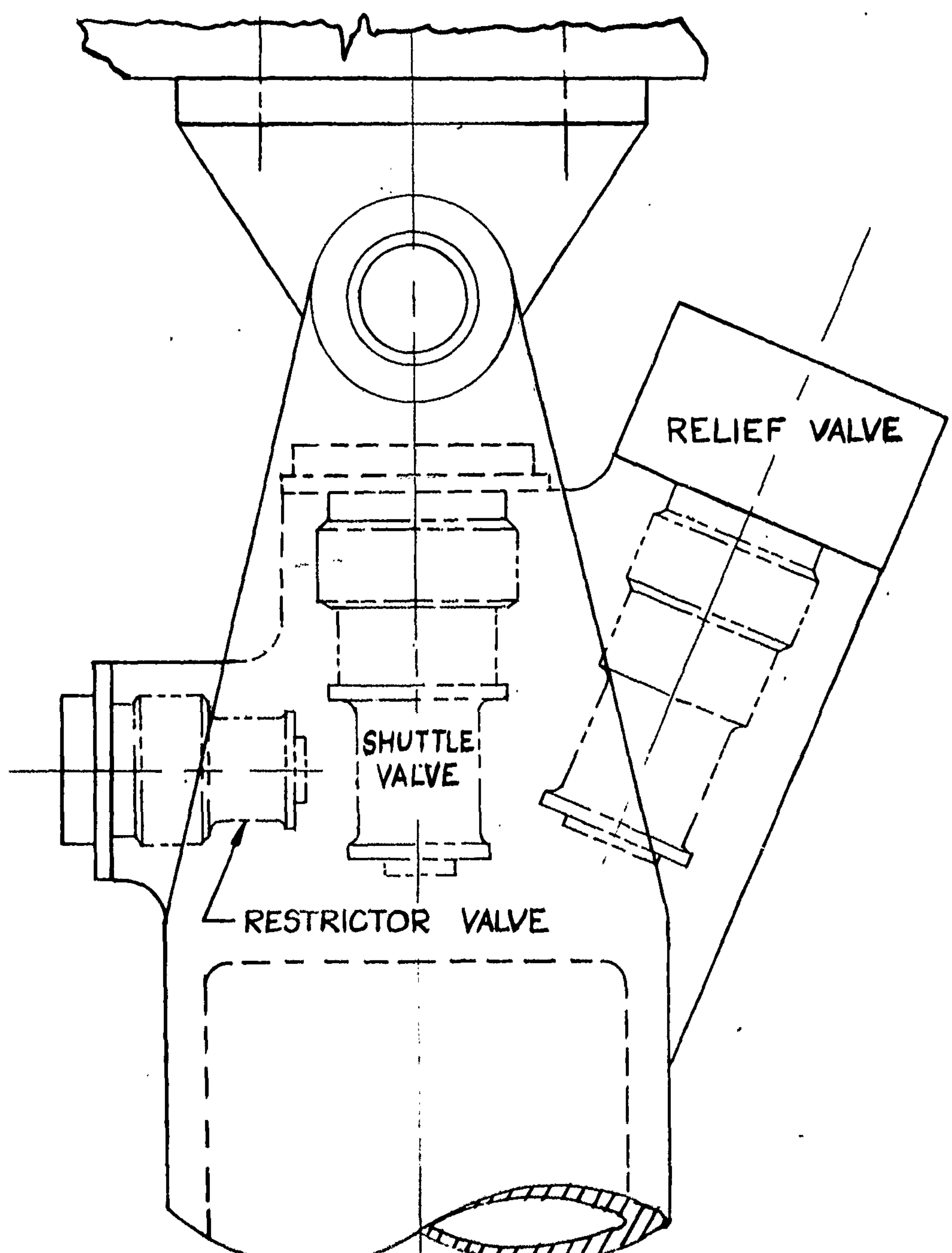


FIGURE III-11

CONFIGURATION "C"

Valves mounted longitudinally around periphery of end cap -- this configuration results in a maximum space requirement but has no effect on the length of the actuator. Maintenance is simplified by removing all valves in the same direction. The design is heavy since material between barrel and valve cavities cannot be machined down to a minimum.

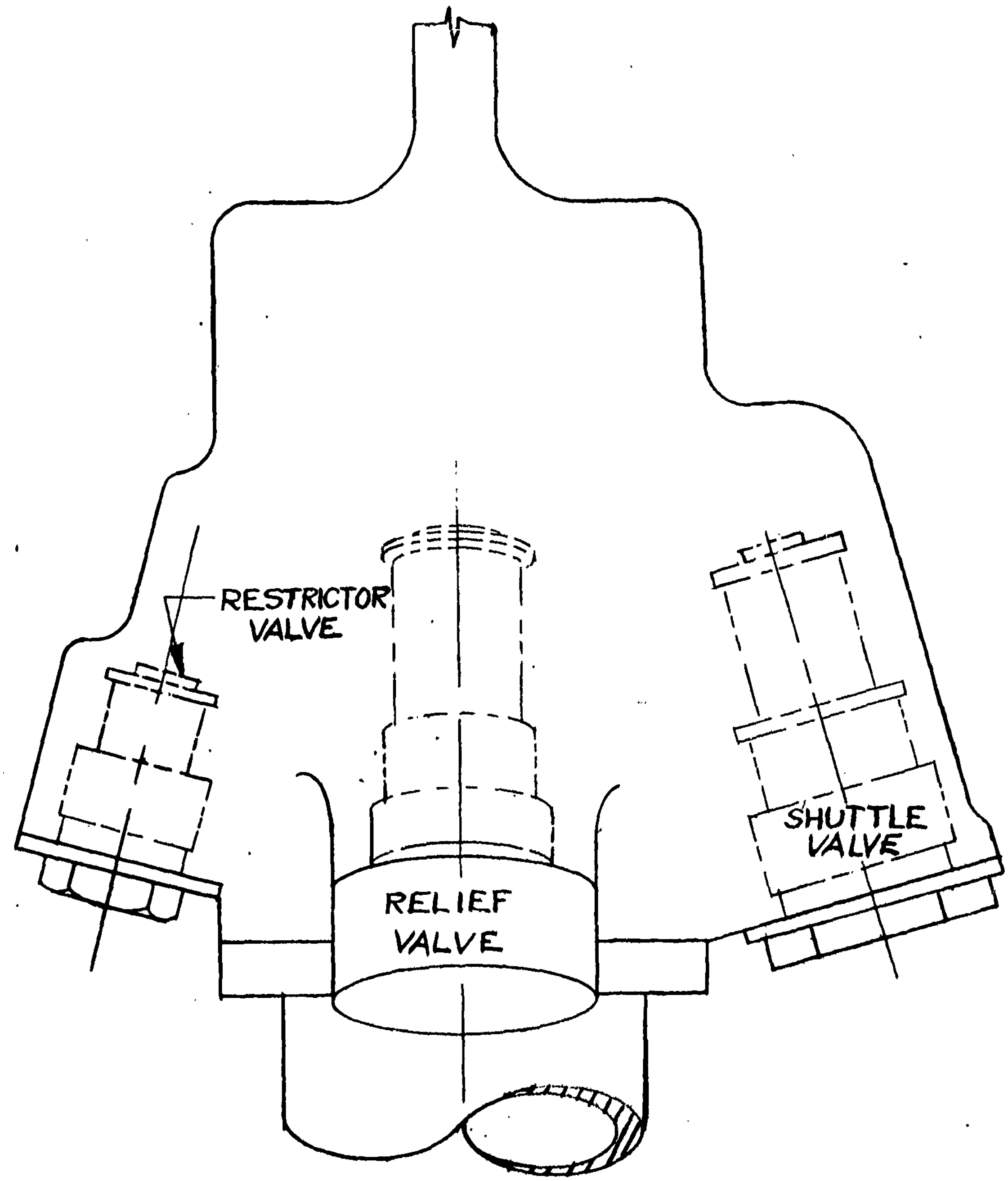
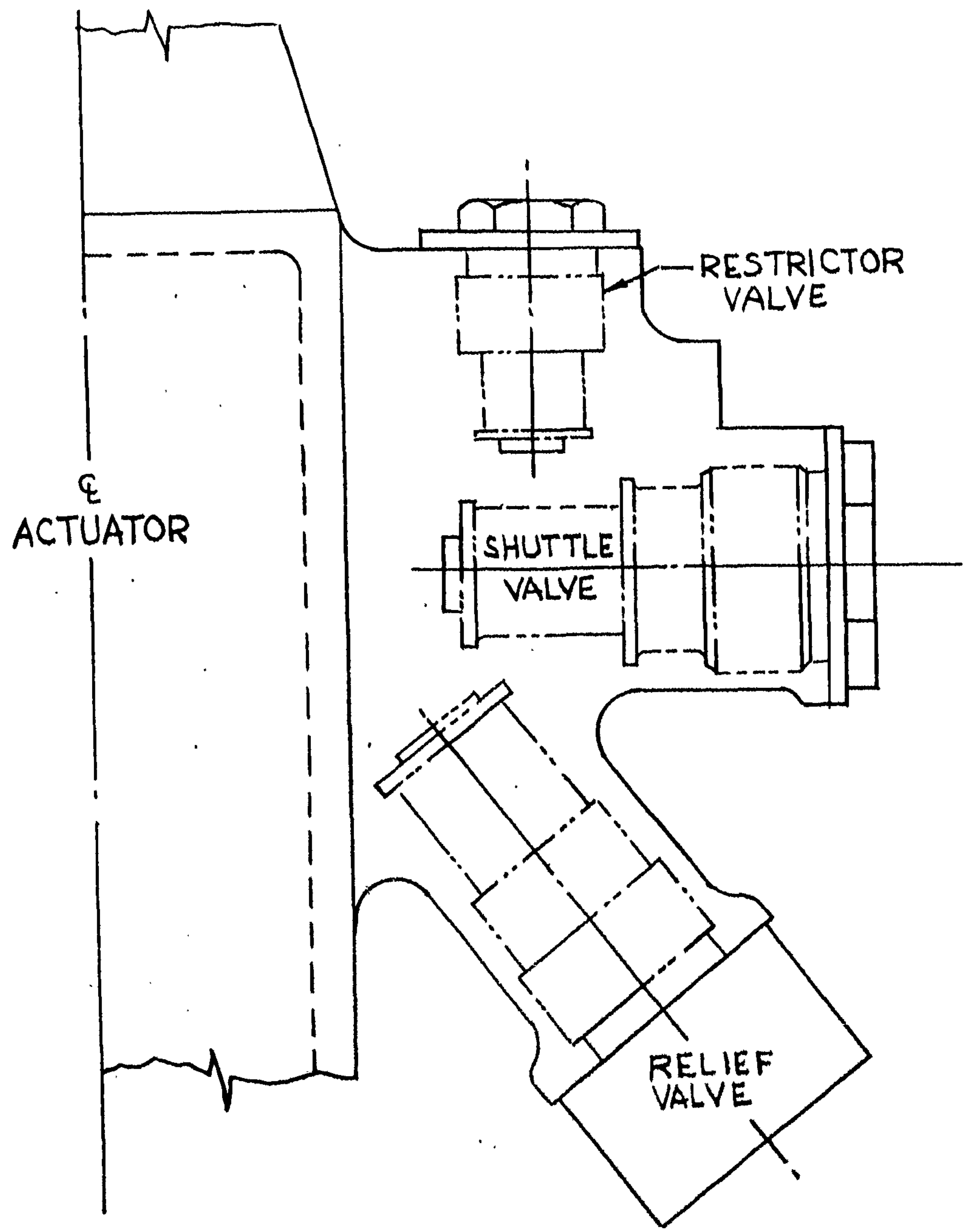


FIGURE III-12

CONFIGURATION "D"

Valves installed in a "block" of material along the side and integral with the actuator barrel -- this configuration requires a maximum of space although the end view is narrow. Maintenance is fair with the valves removable from the same general area. Weight cannot be readily removed from around the valves, making the design very heavy.



set, then when cooled to -20°F and with pressures below 500 psi they would leak. As pressure built up at -20°F, say between 1,000 and 4,000 psi, leakage would cease. For the purpose of performing tests in accordance with Chance Vought specification AER-AVO-53720-0-182 (see Appendix III-2), the Viton seals were very satisfactory.

The results of the tests performed on package #2 in accordance with specification AER-AVO-53720-0-182 are included in Appendix III-6. Some trouble was experienced with the relief valve operation; however, this valve performed in the same manner before being subjected to the package test. It was concluded that the interaction of the valves in this package had no effect on each other's performance.

One failure that occurred when testing #2 package was the inability to affect a seal between the bolted on end cap and the barrel at pressures above 5000 psi. A Hi-Ceal is used in this application, as shown on Chance Vought drawing CVS-54542. The joint sealed at 4000 psi, began to seep at 5000 psi and leaked badly at 6000 psi. This failure was attributed to slight separation of the two cavity sealing faces under high pressure and the inability of the Hi-Ceal to tolerate this movement. From Hi-Ceal tests it was observed that a high mechanical force between the Hi-Ceal and the cavity seats resulted in more reliable sealing. Therefore, from Hi-Ceal springback curves it can be seen that a slight separation of the cavity seats, even .0005 inches, will reduce the force between the Hi-Ceal and cavity seat to approximately one-half the original. This force reduction is enough to cause leakage problems.

From a strength standpoint the joint possessed adequate strength to withstand the required 10,000 psi burst pressure. However, not enough consideration was given to eliminating the deflection between the two cavity sealing surfaces. Because the Hi-Ceal can not tolerate seat movement, deflection becomes the critical design criteria in a bolted together face joint; and, to eliminate deflections the parts must be beefed up beyond that required for stress considerations. Indeed, when sealing is required between bolted together faces where high loads result from proof and impulse pressures of 6000 psi, the elimination of deflection between the cavity seats will be difficult. The deflection not only reduces sealing reliability but during pressure impulse cycling may cause a Hi-Ceal fatigue failure (see section II). The CVS-55240 ball joint actuator design (See Part III) utilized a screw in type end cap with a Hi-Ceal seal. Leakage at this joint was zero. It is also pointed out that sealing of the screw in type modular components was very satisfactory. Based on the above considerations it is recommended that the designer use screw in type end cap installations rather than bolt together types whenever possible.

Another disturbing result attributed to this design is the somewhat high pressure drops measured in the free flow direction. Test results show that 4 GPM flow from C_1 to C_2 (refer to Figure 2 in AER-AVO-53720-0-182) resulted in pressure drops of 1,180 psi at -20°F, 470 psi at 90°F and 240 psi at 450°F. These pressure drops are attributed to the complex flow direction the oil must take to get from C_1 to C_2. Unless the designer takes care in the initial design of the manifold, he is likely to end up with more pressure drop than the system can tolerate.

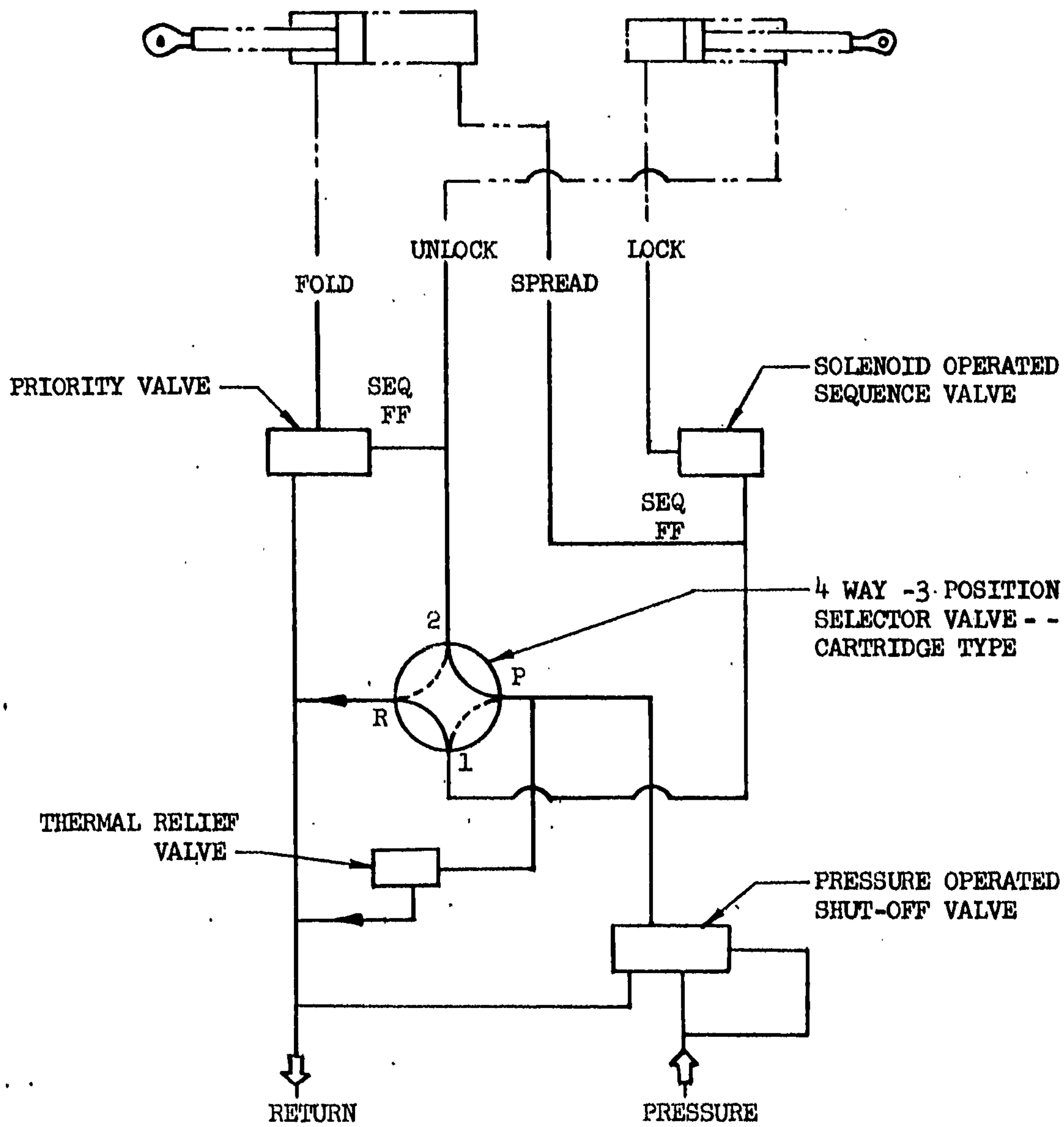
Package #3 - This package was originally planned to simulate a landing gear type actuator. The package was to consist of a Class B shuttle valve, a priority valve, and a face-mounted control valve in conjunction with a ball-joint type actuator. This package was cancelled because the ball joint type actuator was being pursued under the "fluid-carrying structure" program (see Part II), and because the other valves could be démonstrated in other packages.

Package #4 - Package #4 was designed to simulate a wing-fold circuit. A schematic of the package is shown in Figure III-13. It will be noted that some liberties were taken in the selection of valves in order to demonstrate more different types and to show a package which utilizes five modular components. Actually the pressure-operated shut-off would not be required and another solenoid-operated sequence would be a better choice than the priority valve for sequencing. Detail design of the manifold is shown in CVS-54926. Pictures of the finished product are shown in Figures III-14 through III-16.

In order to simulate an airframe design condition, this package was designed to fit into a rather confined space envelope that is defined by the system demonstrator frame and the proposed tube routing. Later in the program the system demonstrator which was part of Phase III was dropped. With the space envelope, the system schematic and required port orientation thus defined for package #4, location of the modular valves for optimum manifold design became a matter of form follows function. The only problems encountered were in refining the design for minimum weight and ease of manufacture. The shape of the modular valves and the fact that five valves were packaged in a "minimum" weight manifold required that the manifold take a rather complex shape. The trend, as observed in the design of packages #2, #4, and #5, is that manifolds will lend themselves more to being cast rather than forged. The complex external shape represents a particularly tough hog-out problem.

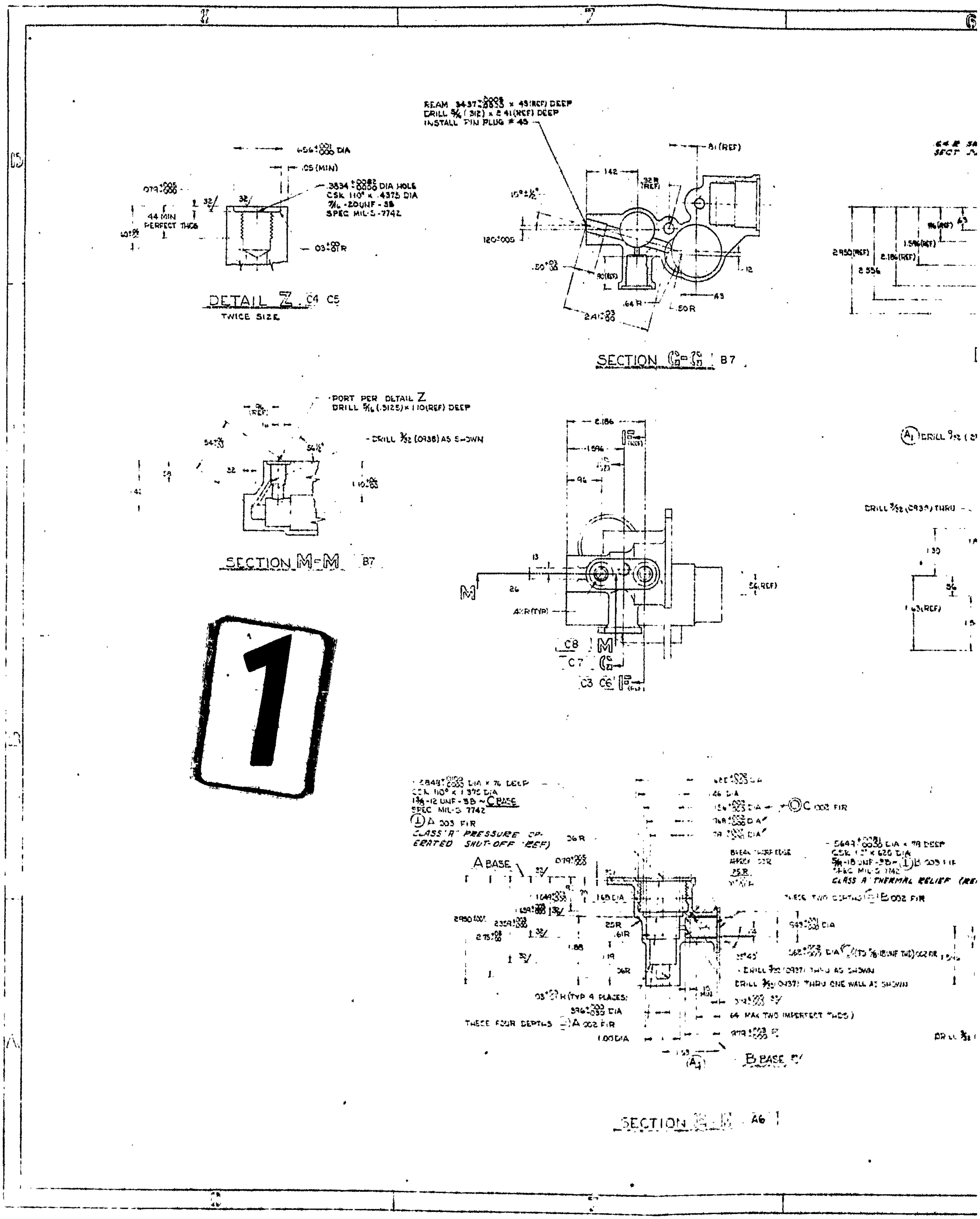
The fabrication process was followed closely by Engineering to gain experience with (a) fabrication problems of 17-4PH material and (b) machining techniques employed on a complex manifold shape. Two simple observations were made: No problems were encountered in the machining of 17-4PH and the cost of machining a complex manifold shape was excessive. Certainly castings should be used for production parts. Refer to page 96 for the discussion on castings and cost.

Package #4 was to be tested in accordance with Chance Vought specification AER-AVO-53720-0-183 (see Appendix III-2). However, due to the cancellation of the pressure-operated shut-off valve, the package was not tested per the test specification. Proof pressure and external leakage tests were performed and were satisfactory.

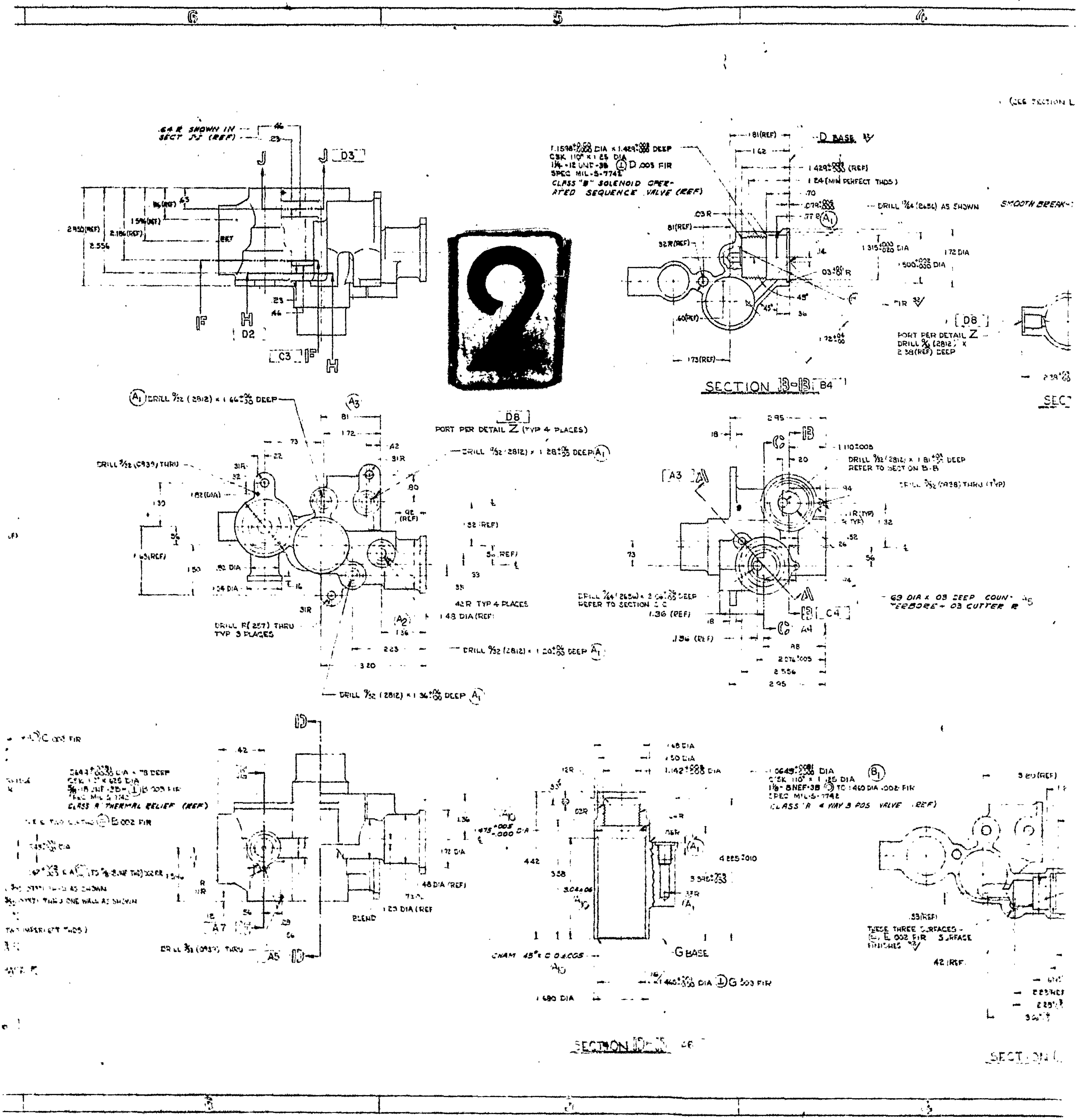


WING FOLD CIRCUIT

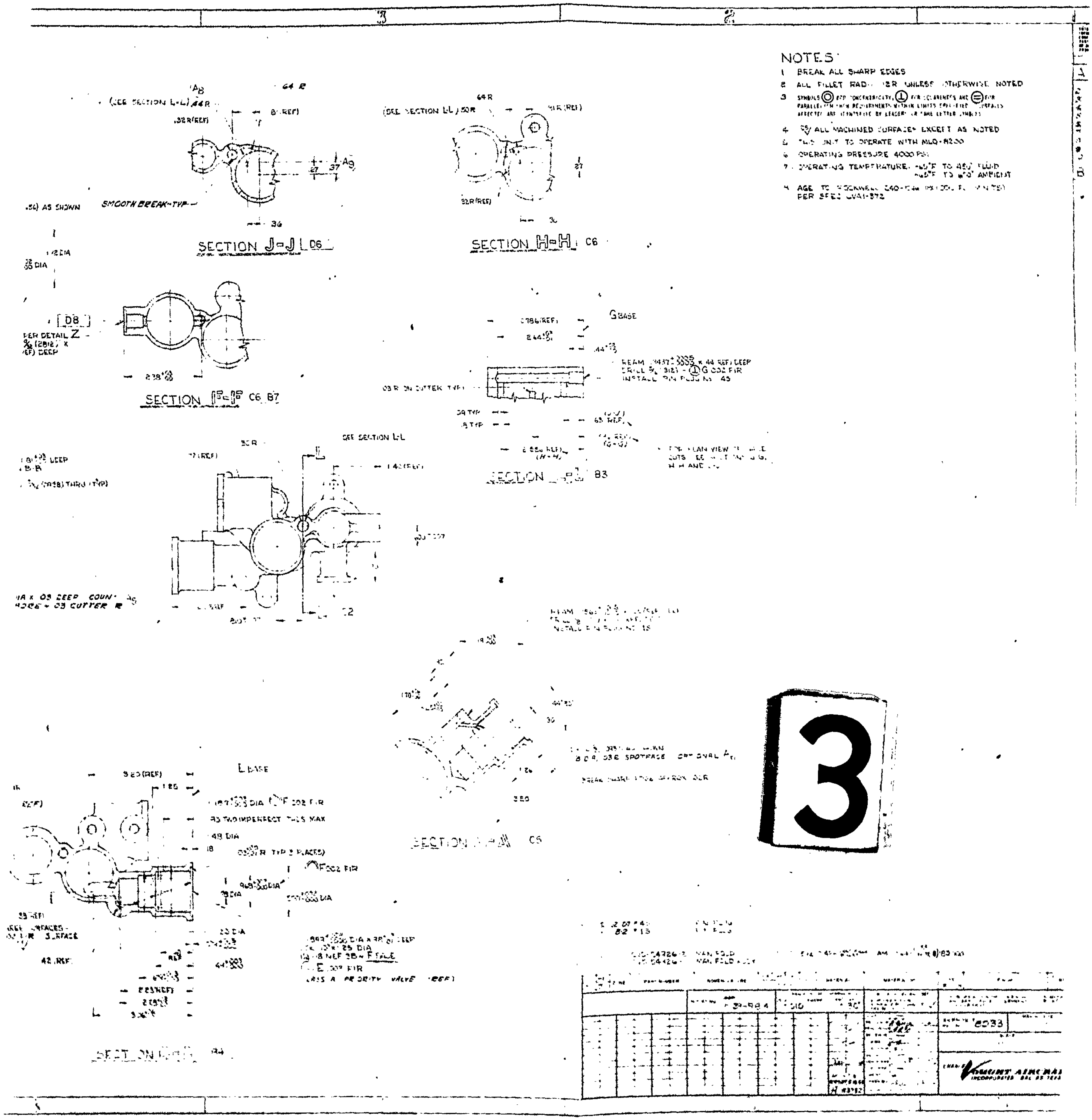
FIGURE III-13
HYDRAULIC SCHEMATIC
PACKAGE #4

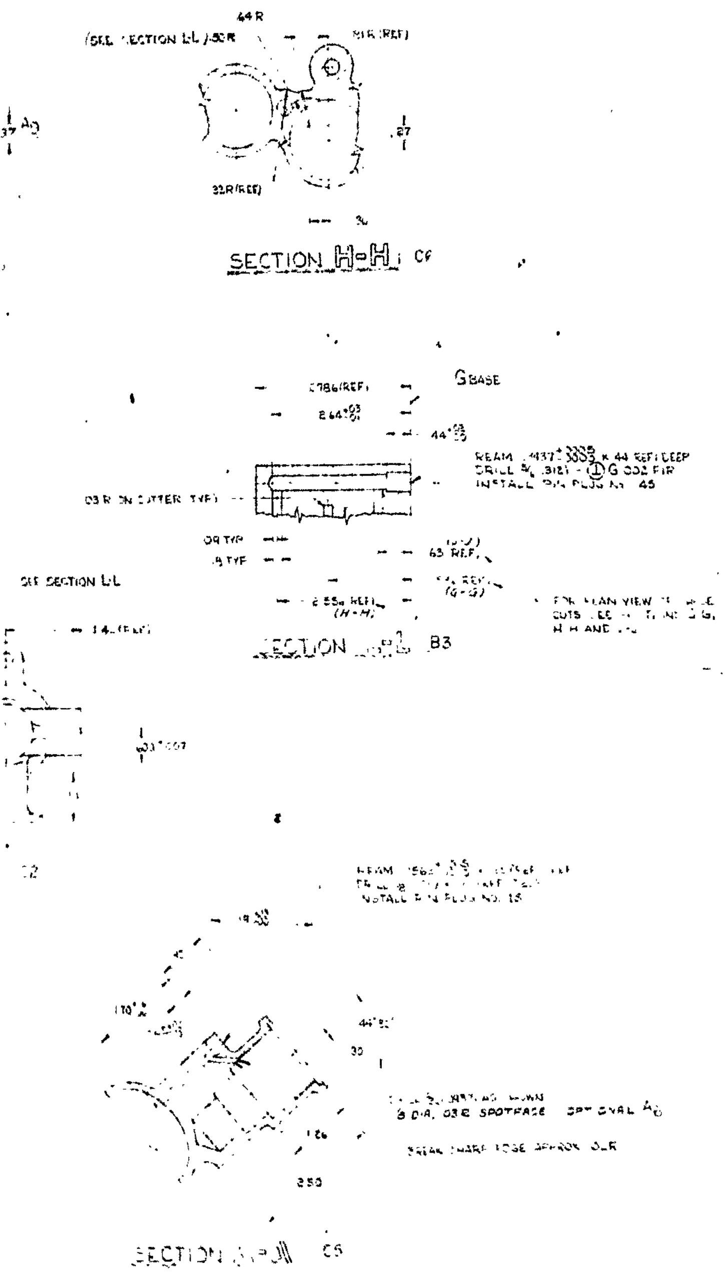


REAM .3437 x .49(REF) DEEP
DRILL 5/16 (.312) x 2.41(REF) DEEP
INSTALL PIN PLUG # 45
.81(REF)
.142
.50R
.64R
.12
DETAIL Z C4 C5
TWICE SIZE
.44 MIN PERFECT THDS
5/16 -20UNF-3B
SPEC MIL-S-7742
SECTION B7
PORT PER DETAIL Z
DRILL 5/16 (.5125) x 1.10(REF) DEEP
DRILL 3/32 (.0938) AS SHOWN
SECTION M-M B7
2.186
1.596
.94
.56(REF)
M
C8
C7
C3 C6
1
CSK. 110° x 1.375 DIA
SPEC MIL-S-7742
CLASS 'A' PRESSURE OPERATED SHUT-OFF (REF)
A BASE
CLASS 'A' THERMAL RELIEF
DRILL 3/32 (.0937) THRU AS SHOWN
DRILL 3/32 (.0937) THRU ONE WALL AS SHOWN
.64 MAX TWO IMPERFECT THDS
THESE FOUR DEPTHS A .002 FIR
1.00 DIA
B BASE
SECTION A6



SECTION B-B B4
PORT PER DETAIL Z (TYP 4 PLACES)
CLASS "B" SOLENOID OPERATED SEQUENCE VALVE (REF)
SMOOTH BREAK
PORT PER DETAIL Z
DRILL 9/32 (2812) THRU (TYP)
42R TYP 4 PLACES
DRILL F (257) THRU TYP 3 PLACES
REFER TO SECTION B-B
REFER TO SECTION C-C
CLASS R THERMAL RELIEF (REF)
CLASS R 4 WAY 3 POS VALVE (REF)
THESE THREE SURFACES
G BASE
D BASE
BLEND
SECTION D-D


NOTES
1 BREAK ALL SHARP EDGES
2 ALL FILLET RAD .12R UNLESS OTHERWISE NOTED
4 ALL MACHINED SURFACES EXCEPT AS NOTED
5 THIS UNIT TO OPERATE WITH MLO-8200
6 OPERATING PRESSURE 4000 PSI
SECTION J-J D6
SECTION H-H C6
SECTION C6 B7
SECTION B3
SECTION C5
SMOOTH BREAK-TYP
SEE SECTION L-L
PER DETAIL Z
03 R ON CUTTER TYP
BREAK SHARP EDGE APPROX .02R
VOUGHT AIRCRAFT
3



NOTES:

1. BREAK ALL SHARP EDGES
2. ALL FILLET RADII .02 R UNLESS OTHERWISE NOTED
3. SYMBOLS ◎ FOR CONCENTRICITY, ⊥ FOR SQUARENESS AND ⊜ FOR PARALLELISM SHOW REQUIREMENTS WITHIN LIMITS SPECIFIED. SURFACES AFFECTED ARE IDENTIFIED BY LEADERS OR SAME LETTER SYMBOLS
4. 63 ALL MACHINED SURFACES EXCEPT AS NOTED
5. THIS UNIT TO OPERATE WITH MLO-8200
6. OPERATING PRESSURE 4000 PSI
7. OPERATING TEMPERATURE: -65°F TO 450° FLUID; -65°F TO 600° AMBIENT
8. AGE TO ROCKWELL C40-C44 (180,000 F... (V N TS) PER SPEC CVAI-372

REVISIONS

A	1. ADDED MISSING DIMENSION D-4, A-5, A-5, B-5, B-5, C-5, C-7
	2. .36 DIMENSION WAS .34 - A-5
	3. .81 DIMENSION WAS (REF) - C-6
	4. .63 DIMENSION WAS (REF) - C-7
	5. .38 CBR ADDED - B-4
	6. 1/8 SPOTFACE ADDED - B-2
	7. CENTERED VALVE CAVITIES
	8. .44 R WAS .50 R (SECT J-J B-2)
	9. .37 DIMENSION WAS .27 (J-J B-2)
	10. ADDED UCUT & CHAM SECT D-D R-5)
B	1. ... WAS ... SECT ...

MANIFOLD ASSY

VOUGHT AIRCRAFT INCORPORATED DALLAS TEXAS

CVS-54926 A

4

NOTE: REFER TO FIGURE II-2 FOR SCHEMATIC OF #4 PACKAGE

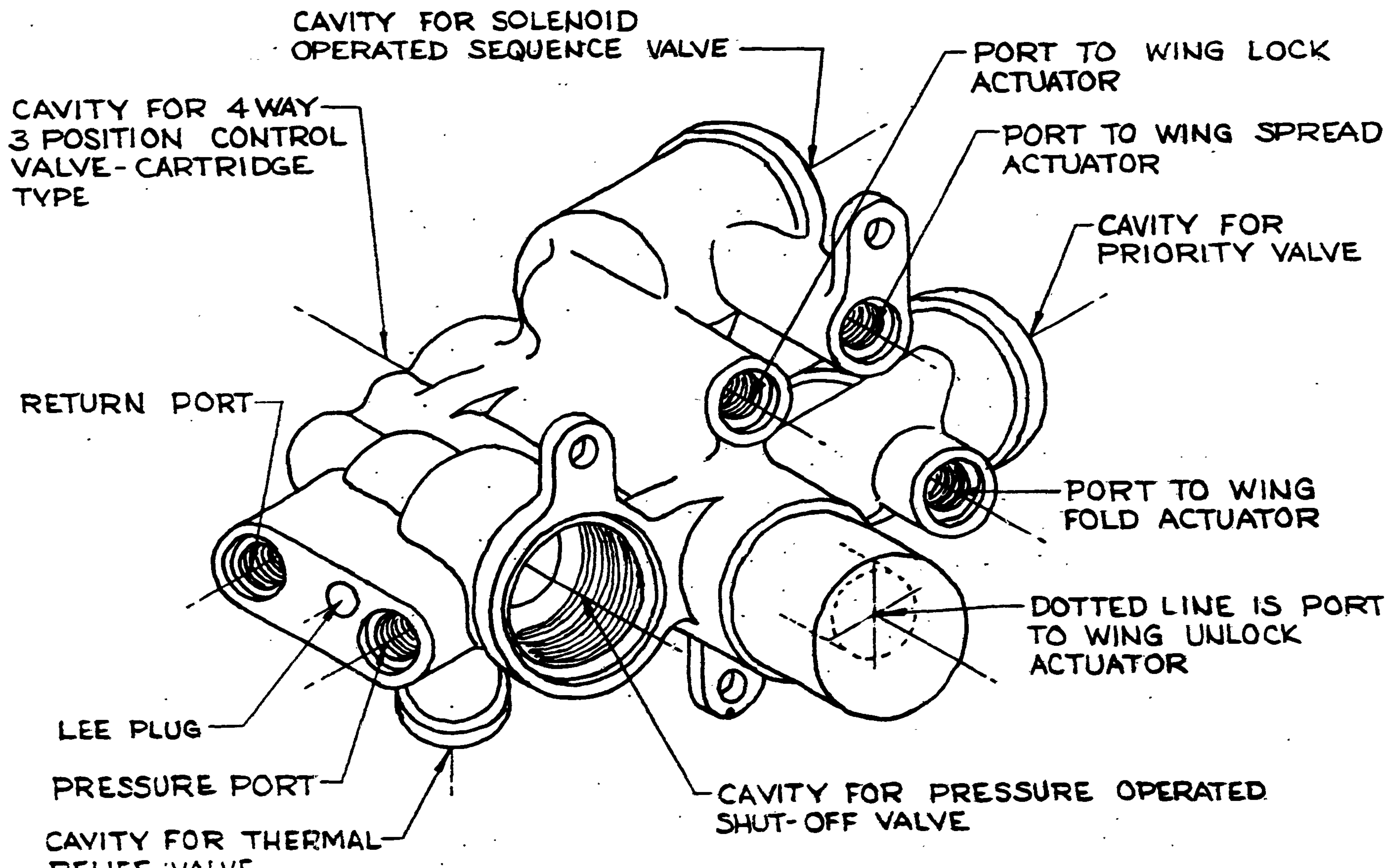
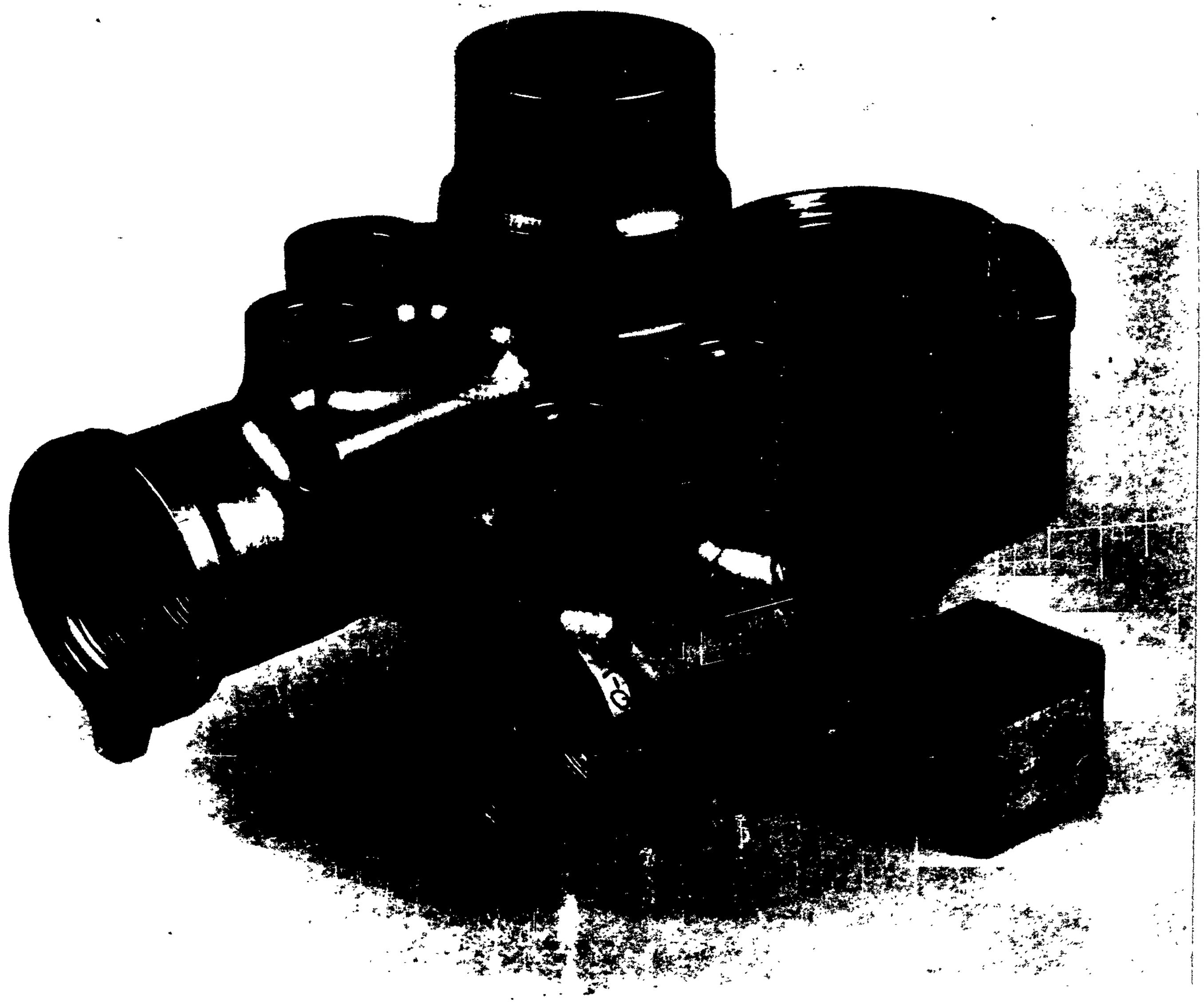
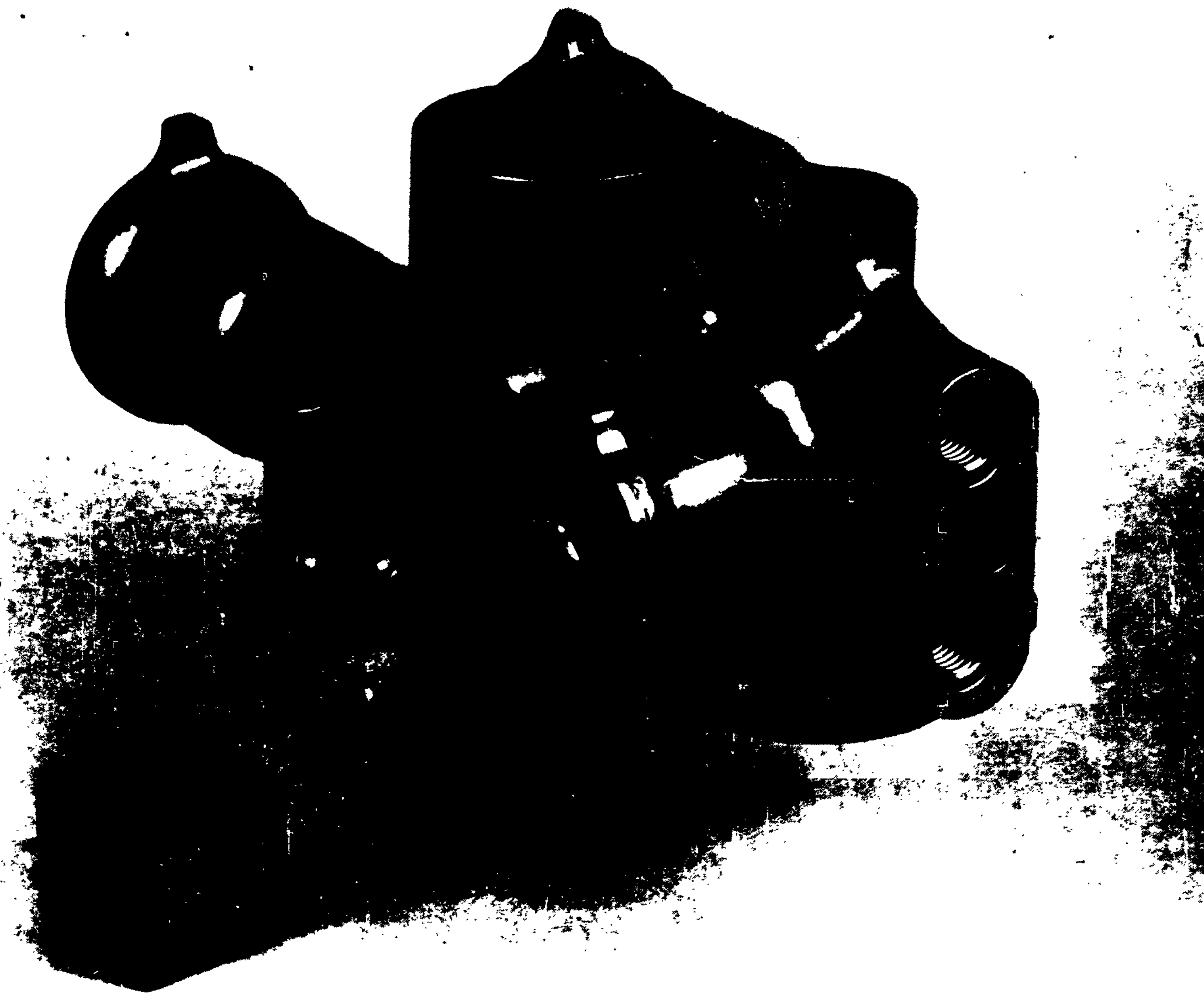


FIGURE III-14

SKETCH OF #4 PACKAGE MANIFOLD
APPROXIMATELY 3/4 SIZE



CVS-54926 MODULAR HYDRAULIC MANIFOLD FOR PACKAGE #4. FOR USE WITH THERMAL RELIEF, PRIORITY, PRESSURE - OPERATED SHUT-OFF, SOLENOID OPERATED



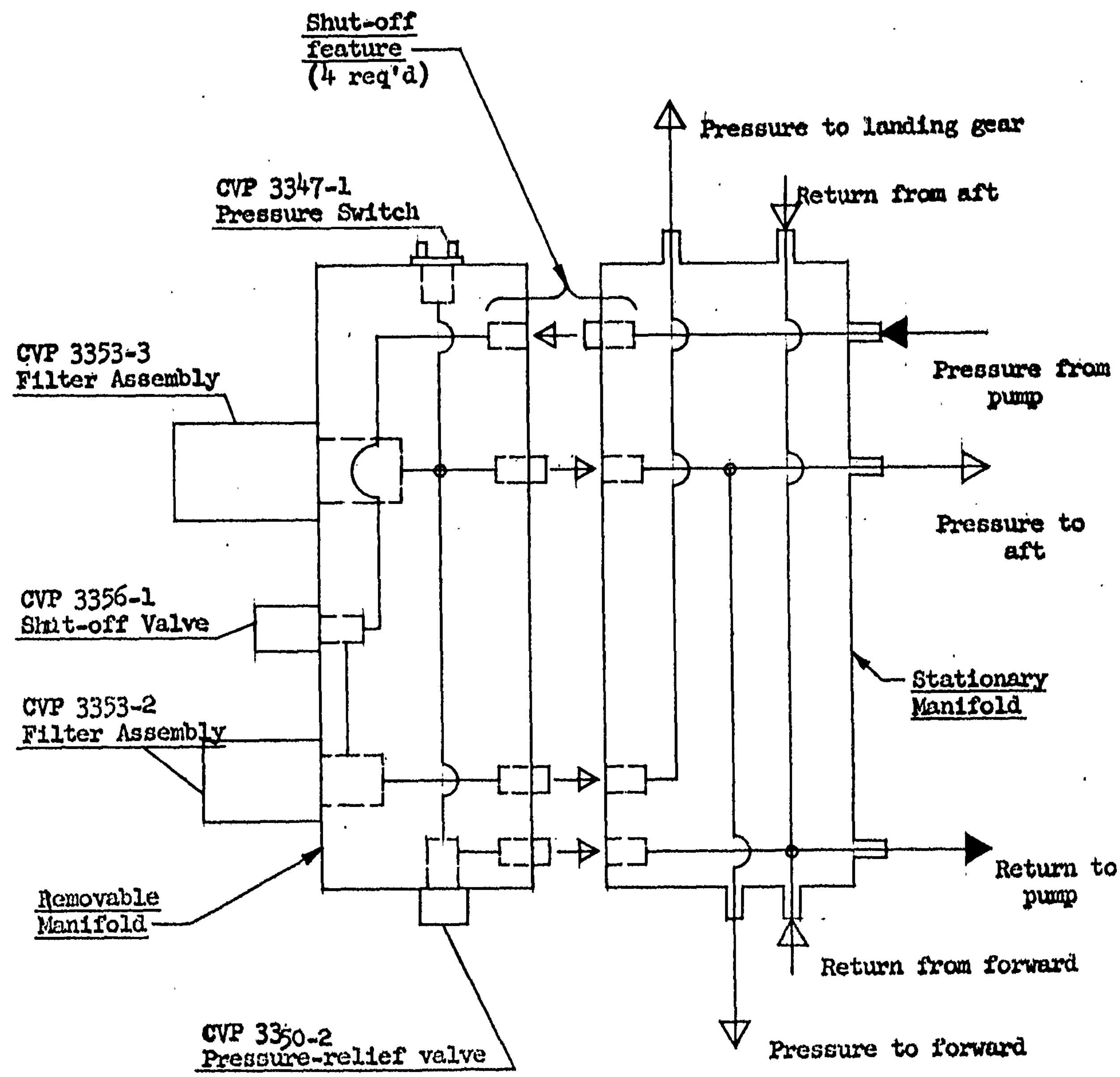
CVS-54926 MODULAR HYDRAULIC MANIFOLD FOR PACKAGE #4. FOR USE WITH THERMAL RELIEF, PRIORITY, PRESSURE - OPERATED SHUT-OFF, SOLENOID OPERATED SEQUENCE AND 4W-3P CONTROL VALVES.

Package #5 - This package consists of two manifolds face-mounted together. When separated, there are four common open ports in each manifold which would normally cause loss of fluid and system shut-down until reassembly. This package employs an automatic shut-off feature in each exposed port, permitting separation of the two manifolds with no appreciable loss in fluid and allowing system operation to continue while the #5 package removable manifold is removed. This design illustrates an approach to package removal without appreciable loss of fluid as well as demonstrating a servicing approach. Standby packages could replace malfunctioning packages.

A schematic of the #5 package is shown in Figure III-17. Detail design of the manifolds are shown in CVS-54943 and CVS-54944. Drawings of the detail parts are included in Appendix III-7. Pictures of the finished product are shown in Figures III-18 through III-23.

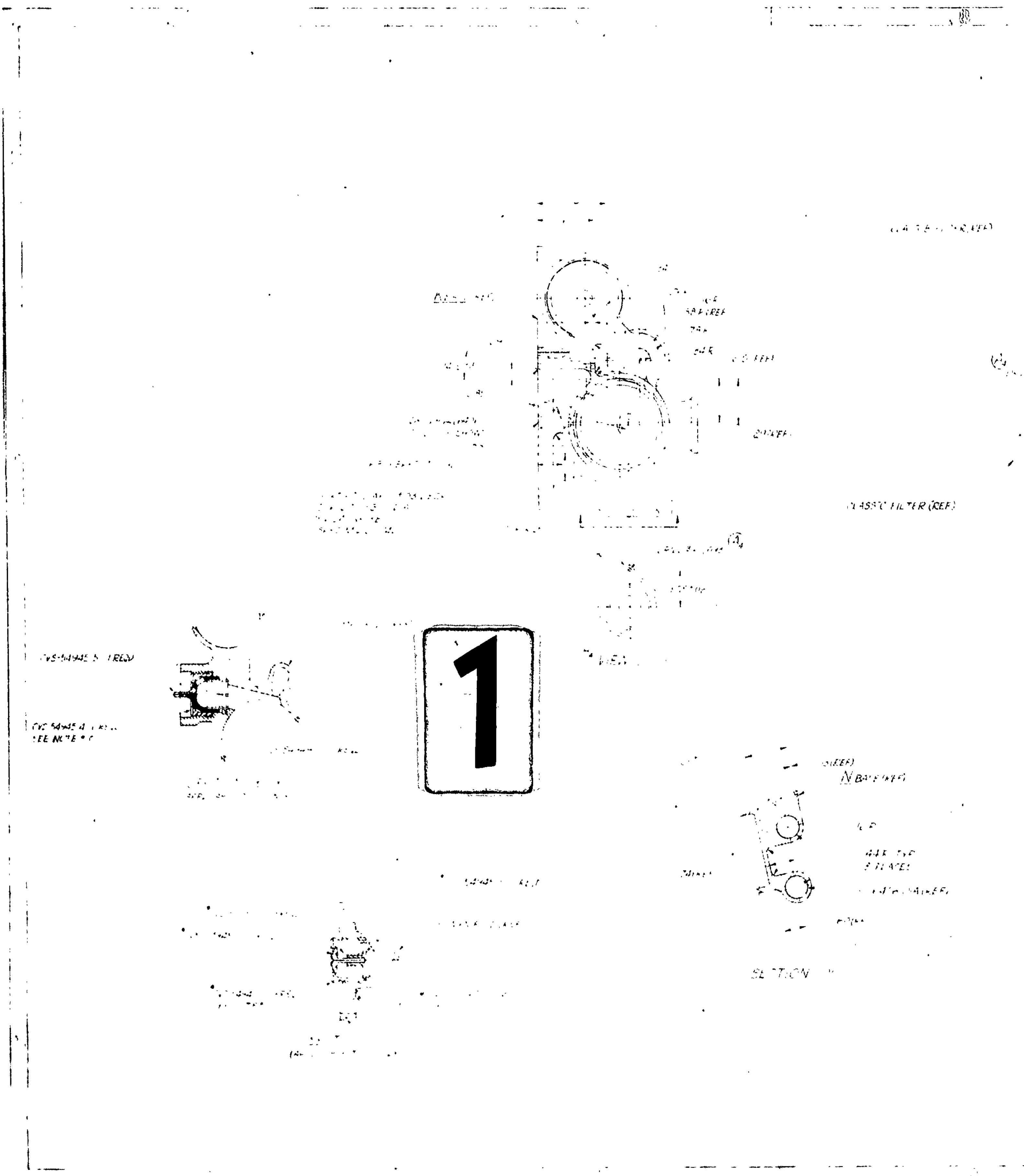
Both manifolds of package #5 definitely lend themselves to casting rather than forgings or machine hog-outs. It is felt that investment castings of the wax or mercury process will be practical for production quantities. Attempts were made to design #5 package in such a way as to readily produce the manifolds by machine hog-out or forging. But in every case, the package became much heavier and less compact than when the configuration was designed as a casting. As in package #4 the trend is that when grouping three or more modular valves a casting is the best approach. Although CVS-54943 was designed as a casting, it was completely machined because of the prohibitive cost of a single casting (see discussion on page 97). The machining job on this package was done by Ellanef Machine Tool Company, 97-11 -- 50th Avenue, Corona 68, New York. All the other manifolds were machined by Chance Vought.

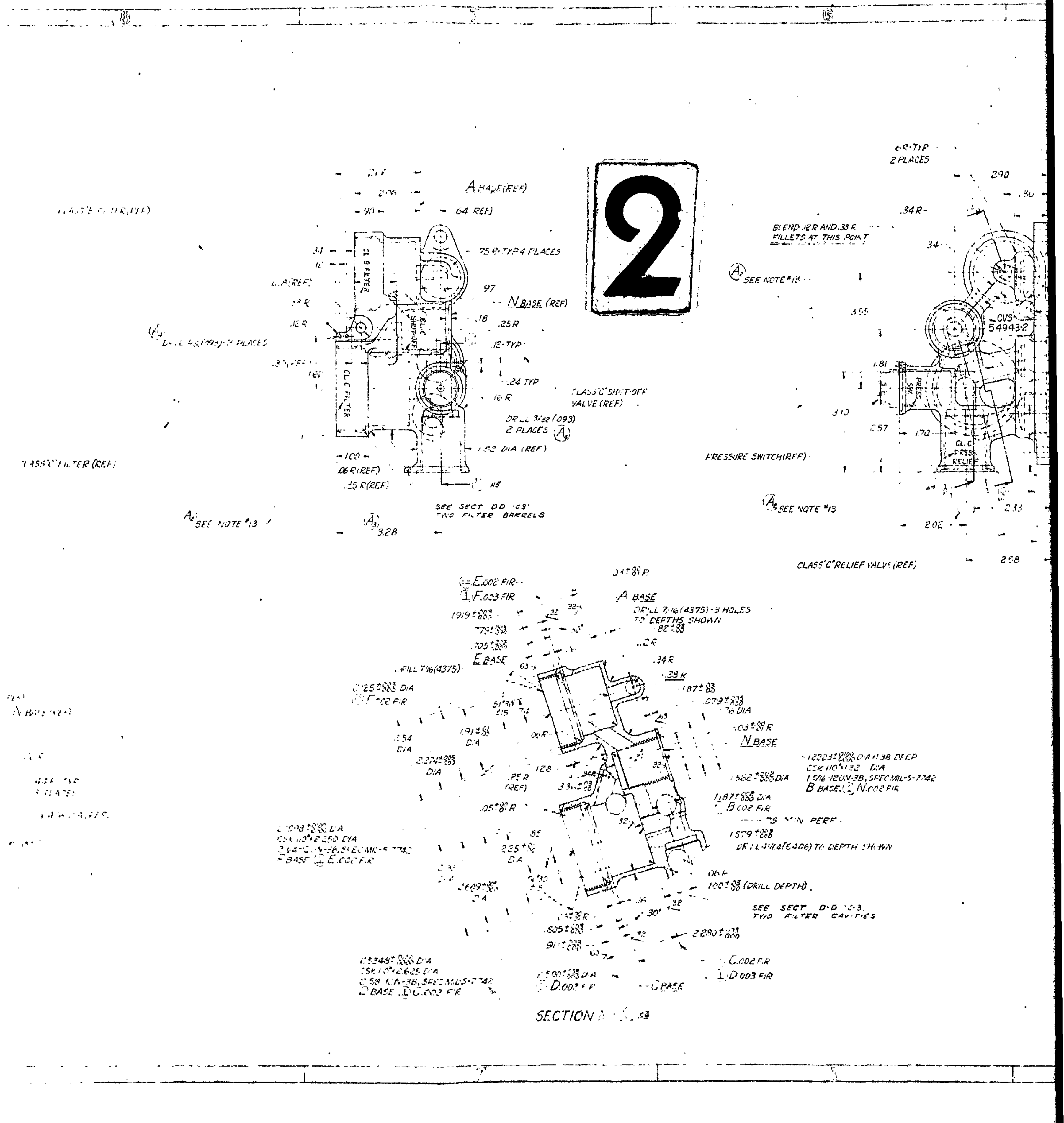
The results of the tests performed on package #5 in accordance with specification AER-AVO-53720-0-184 (see Appendix III-2) are included in Appendix III-6. One outstanding deficiency of the design was noted when pressure was applied to the complete package. Leakage occurred at the seals between the two manifolds. With the bolts across this joint torqued to their maximum value, seepage occurred at 500 psi and leakage was profuse at 2,000 psi. The Hi-Ceals were replaced with elastomer O-rings and there was no leakage when pressure was applied. However, measurements showed that the faces were parting as much as 0.004 inches with 4,000 psi applied to the manifolds and recommended torque applied to the bolts. Since it has been learned that metallic seals cannot tolerate movement of the seal seating surfaces, both from a fatigue and leakage standpoint, it must be concluded that the design was inadequate. The tests were completed on package #5 with O-rings replacing the metallic seals between the two manifolds.

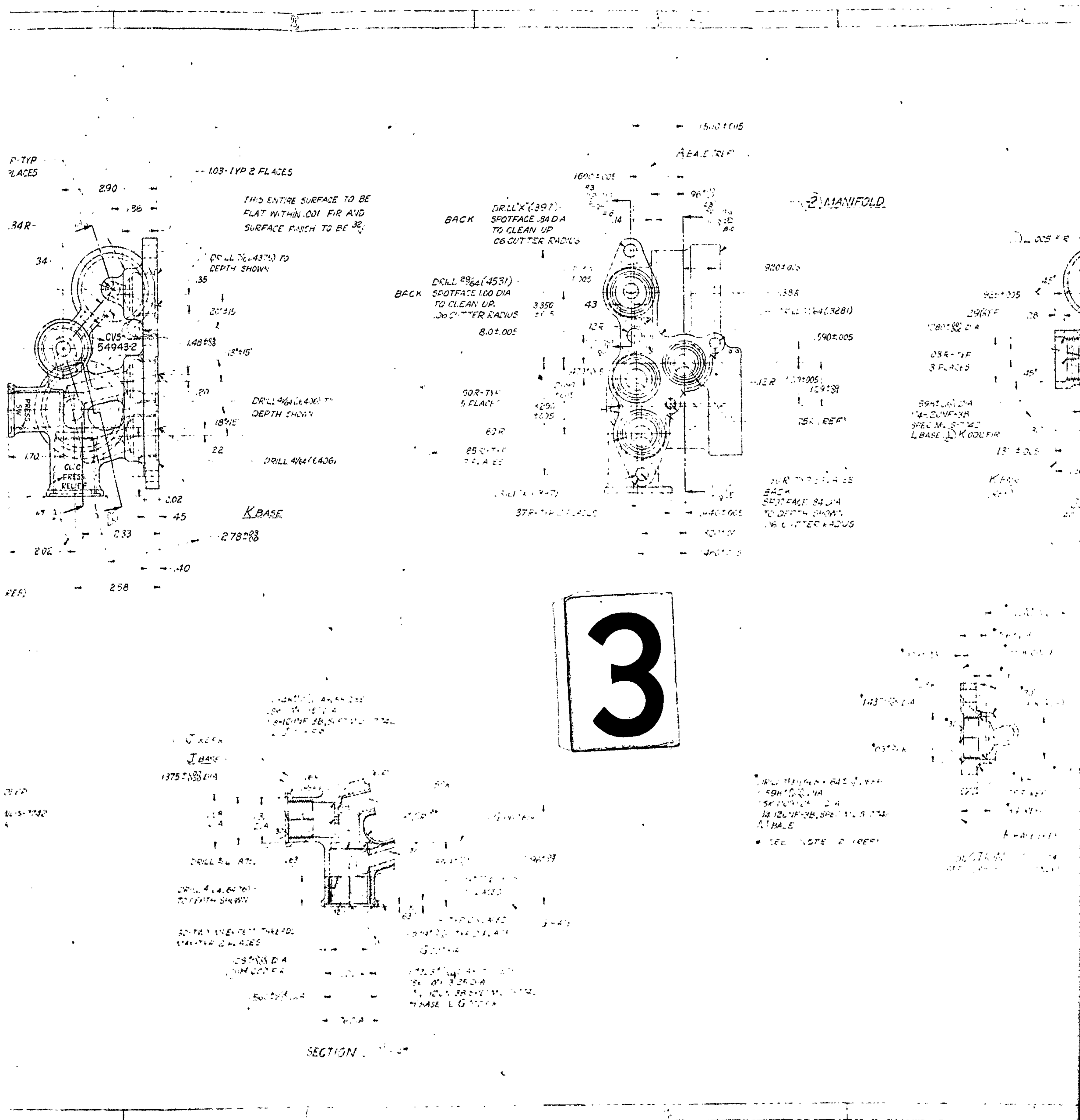


#5 PACKAGE SCHEMATIC

FIGURE III-17


1


2
A BASE (REF)
.64 (REF)
.75 R TYP 4 PLACES
N BASE (REF)
.25 R
.12 TYP
.24 TYP
.16 R
CLASS 'C' SHUT-OFF VALVE (REF)
DRILL 3/32 (.093) 2 PLACES
.152 DIA (REF)
.06 R (REF)
.25 R (REF)
SEE SECT DD
TWO FILTER BARRELS
SEE NOTE #13
3.28
CLASS 'C' FILTER (REF)
CL B FILTER
CL C FILTER
6R TYP 2 PLACES
2.90
.34 R
BLEND .12 R AND .38 R FILLETS AT THIS POINT
SEE NOTE #13
CVS 54943-2
3.55
1.81
2.57
.170
CL C PRESS RELIEF
PRESS SW
FRESSURE SWITCH (REF)
SEE NOTE #13
2.33
2.02
2.58
CLASS 'C' RELIEF VALVE (REF)
E .002 FIR
F .003 FIR
A BASE
DRILL 7/16 (.4375) - 3 HOLES TO DEPTHS SHOWN
E BASE
DRILL 7/16 (.4375)
.34 R
.38 R
N BASE
.128
.25 R (REF)
B .002 FIR
.75 MIN PERF
DRILL 41/64 (.6406) TO DEPTH SHOWN
.100 (DRILL DEPTH)
SEE SECT D-D
TWO FILTER CAVITIES
C .002 FIR
D .003 FIR
C BASE
SECTION



1.03-TYP 2 PLACES
2.90
THIS ENTIRE SURFACE TO BE FLAT WITHIN .001 FIR AND SURFACE FINISH TO BE 32
DRILL 7/16 (.4375) TO DEPTH SHOWN
CV5-54943-2
PRESS SW
CHECK PRESS RELIEF
DRILL 26/64 (.406) TO DEPTH SHOWN
DRILL 26/64 (.406)
K BASE
2.78
2.58
2.33
2.02
BACK
DRILL "X" (.397) SPOTFACE .84 DIA TO CLEAN UP .06 CUTTER RADIUS
DRILL 29/64 (.4531) SPOTFACE 1.00 DIA TO CLEAN UP .06 CUTTER RADIUS
8.0±.005
A BASE (REF)
2 MANIFOLD
BACK SPOTFACE .84 DIA TO DEPTH SHOWN .06 CUTTER RADIUS
J BASE
SECTION
3

NOTES:

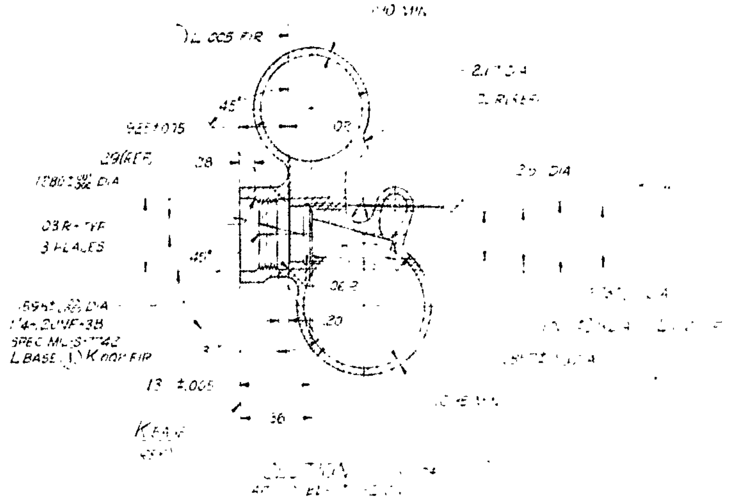
1. UNLESS OTHERWISE NOTED:

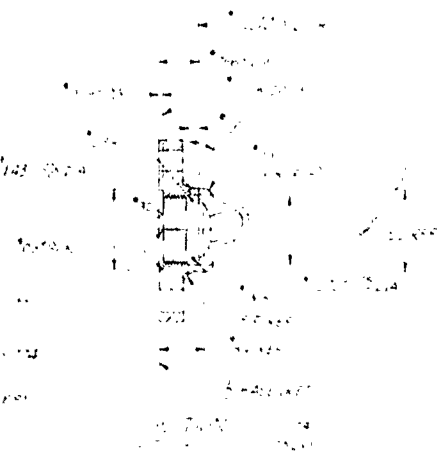
OPERATING PRESSURE

REVISIONS

A 1 ADDED NOTE #11, #12
2 ADDED NOTE #13, IDENTIFICATION AND PART NUMBER
3 ADDED "3.28"
ADDED DRILL 3 PLACES

FOLD





4

MANIFOLD ASSY
MODULAR HYDRAULICS
PACKAGE #5 - REMOVABLE

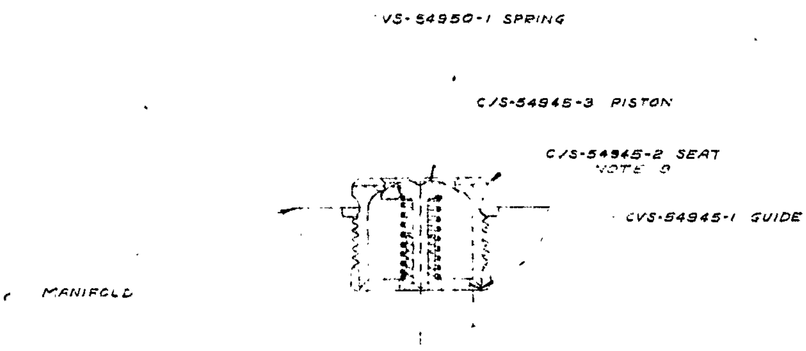
FULL

CHANCE VOUGHT AIRCRAFT INCORPORATED DALLAS TEXAS

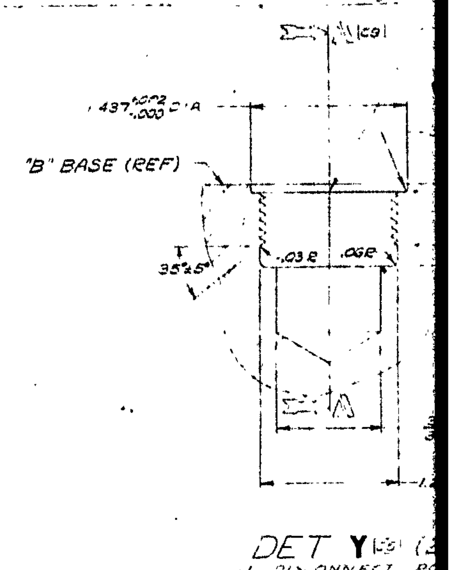
CVS-54943

A

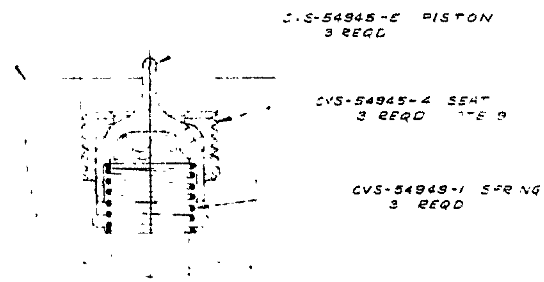
CVS-54944 33xE



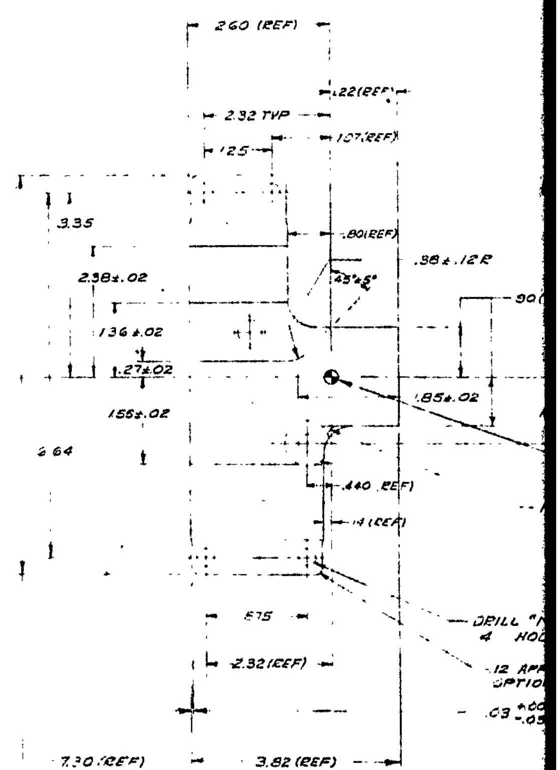
SECT A – A (2X)
APPLICABLE -1 ASSY ONLY



DET Y (2
(1 DISCONNECT PO

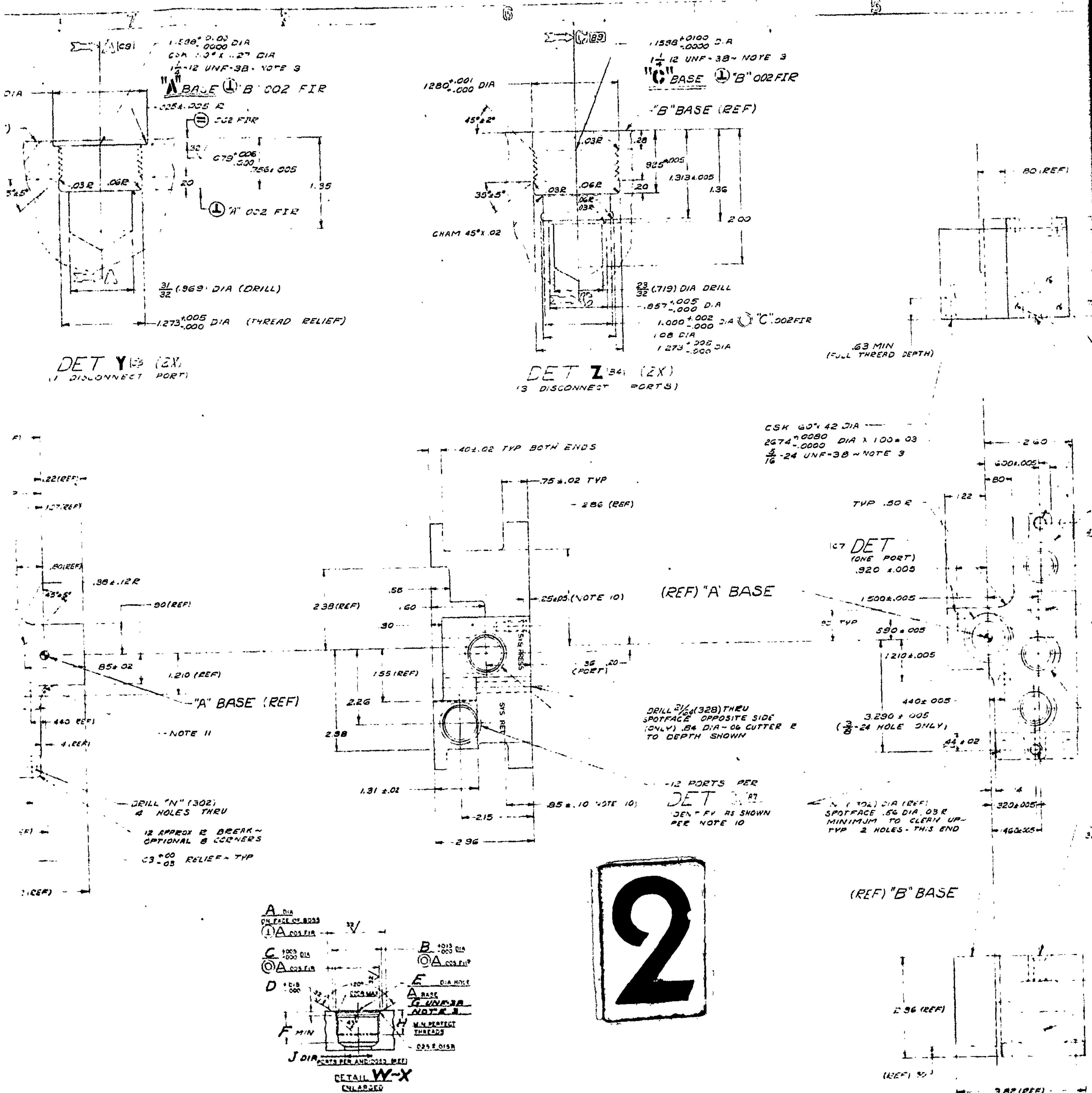


SECT C – C (2X)
APPLICABLE -1 ASSY ONLY



1

-1 MANIFOLD ASSY
ASSEM PER NOTE 9



DET	DASH No	A DIA	B DIA	C DIA	D DEPTH	E DIA ± TOLERANCE	F DEPTH	G THREAD	H DEPTH	J DIA & DEPTH
W	-10	1.16	1.020	.895	.107	.7077 + .0031	.78	$\frac{7}{8}$-14	.63	$\frac{51}{64}$ (.797) x 1.15
X	-12	1.41	1.234	1.086	.125	.8723 + .0100	.91	$1\frac{1}{16}$-12	.69	$\frac{31}{32}$ (.968) x .34

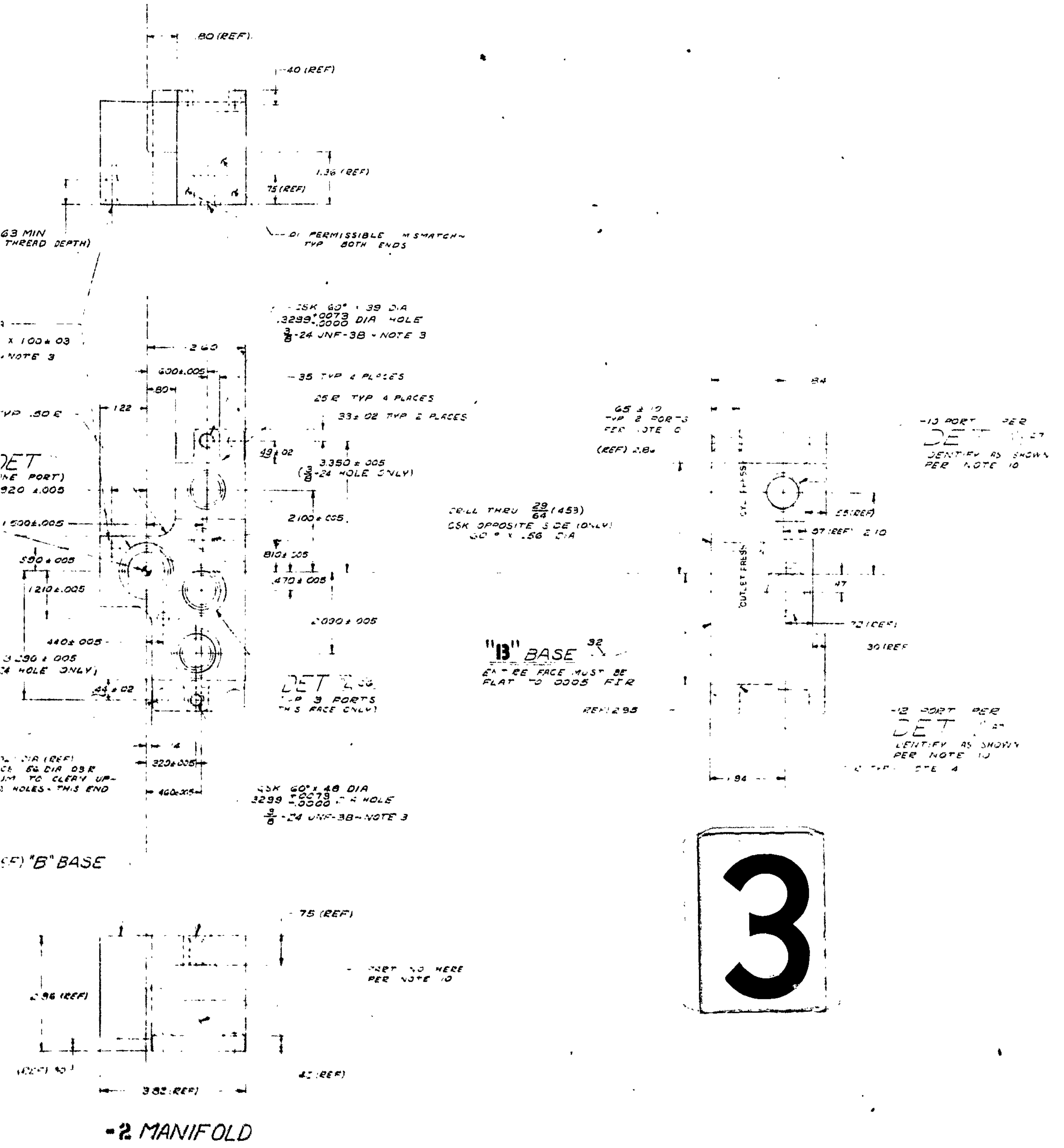
-2 MANIFOLD

ENGINEERING N

1 OPERATING FLUID: ... CHEMICAL CO ... SAN FRA...

2 OPERATING TEMPERAT... FLUID ... AMBIENT ...

3 OPERATING PRESSURES ... PROOF ... BURST ...



3

CVS 54950-1
3 88 CVS-54949-1
CVS 54945
1 08 CVS 54945
CVS-54944-2
CVS-54944-1

ENGINEERING INFORMATION:

1 OPERATING FLUID: MLO 8200 ORONITE CHEMICAL CO. SAN FRANCISCO CALIF)

2 OPERATING TEMPERATURES, F°:
FLUID -65 +450
AMBIENT -65 +650

3 OPERATING PRESSURES PSI:
NORMAL 4000
PROOF & SURGE 6000
BURST 10 000

NOTES:

1 FINISH UNLESS OTHERWISE SPECIFIED
25 TURNED OR BORED SURFACES
DRILLED HOLES
250 EXTERNAL MILLED SURFACES

2 BREAK ALL SHARP EDGES (SEE NOTE 7)

3 THREADS PER SPEC MIL-S-7742

4 ALL FILLETS .12 R UNLESS SPECIFIED

5 [illegible]

6 HEAT TREAT TO ROCKWELL C40-44 (180,000 PSI MIN UTS) PER SPEC [illegible]

7 VAPOR HONE ALL INTERIOR SURFACES TO REMOVE BURRS LOOSE EDGES ETC WHICH MIGHT BECOME DISLODGED DUE TO FLUID FLOW. HONING EXTERNAL SURFACES OPTIONAL.

8 MAGNETIC PARTICLE INSPECT PER MIL-I-[illegible]

9 REFER TO ASSEMBLY OF CVS 54944- [illegible] MODULAR HYDRAULICS UNIT 53732 [illegible] FOR LOCKTITE SEALANT & ASSEMBLY INFORMATION

10 IDENTIFY PORTS & PART NO BY ENGRAVING PER SPEC CVR 5-530 OR BY PERMANENT MARKING WITH VIBRATING PENCIL PER SPEC CVR 9-209 FIGURE HEIGHT .12 [illegible]

11 CLEAN UP OUTLINED AREA SURROUNDING SPOTFACE OR COUNTERBORE TO .025 - .015 ABOVE FLUSH WITH FLAT DIAMETER .250 PERMISSIBLE. USE CUTTER WITH SAME CORNER RADIUS AS SPOTFACE. (ZONE B-7

REVISIONS

-10 PORT PER
DET [illegible]
IDENTIFY AS SHOWN
PER NOTE 10

(REF)
(REF) 2.10
(REF)
.30 (REF)

-2 PORT PER
DET [illegible]
IDENTIFY AS SHOWN
PER NOTE 10

4

REF CVS 5499 [illegible] TORQUE TOOL [illegible] NOTE 9

1 09 CVS-54950- [illegible]
3 88 CVS-54949- [illegible]

1 08 CVS-54945 [illegible]
3 08 -4 [illegible]
1 08 -3 [illegible]
1 08 -2 [illegible]
1 08 CVS 54345-1 [illegible]

1 09 CVS-54944-2 MANIFOLD 3 4 7/8 [illegible] 3 5343 [illegible]
A9 CVS-54344-1 TEST [illegible]

53979-5 ±.010 NOTED ±30°

8033 34-89

FULL

MANIFOLD ASSY
MODULAR HYDRAULICS
PKG 5 STATIONARY

SCHMIDT 4-28-60
54232

VOUGHT AIRCRAFT

CVS-54944

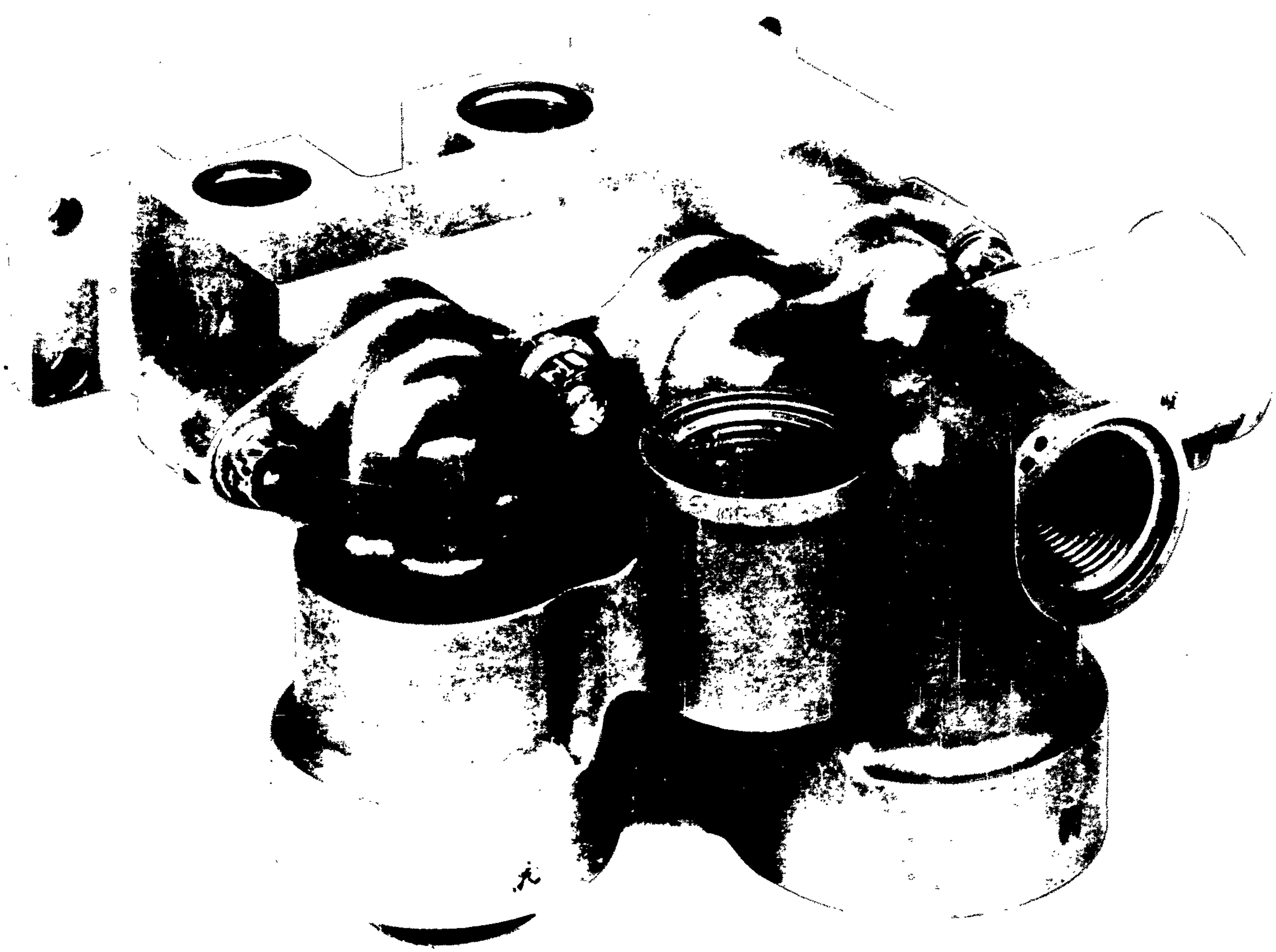


FIGURE III-18

CVS-54943 AND CVS-54944 MODULAR HYDRAULIC MANIFOLDS FOR PACKAGE #5. FOR USE WITH CLASS B AND CLASS C FILTERS, SOLENOID - OPERATED SHUT-OFF VALVE, PRESSURE RELIEF VALVE AND PRESSURE SWITCH. UPPER LEFT-HAND VIEW.

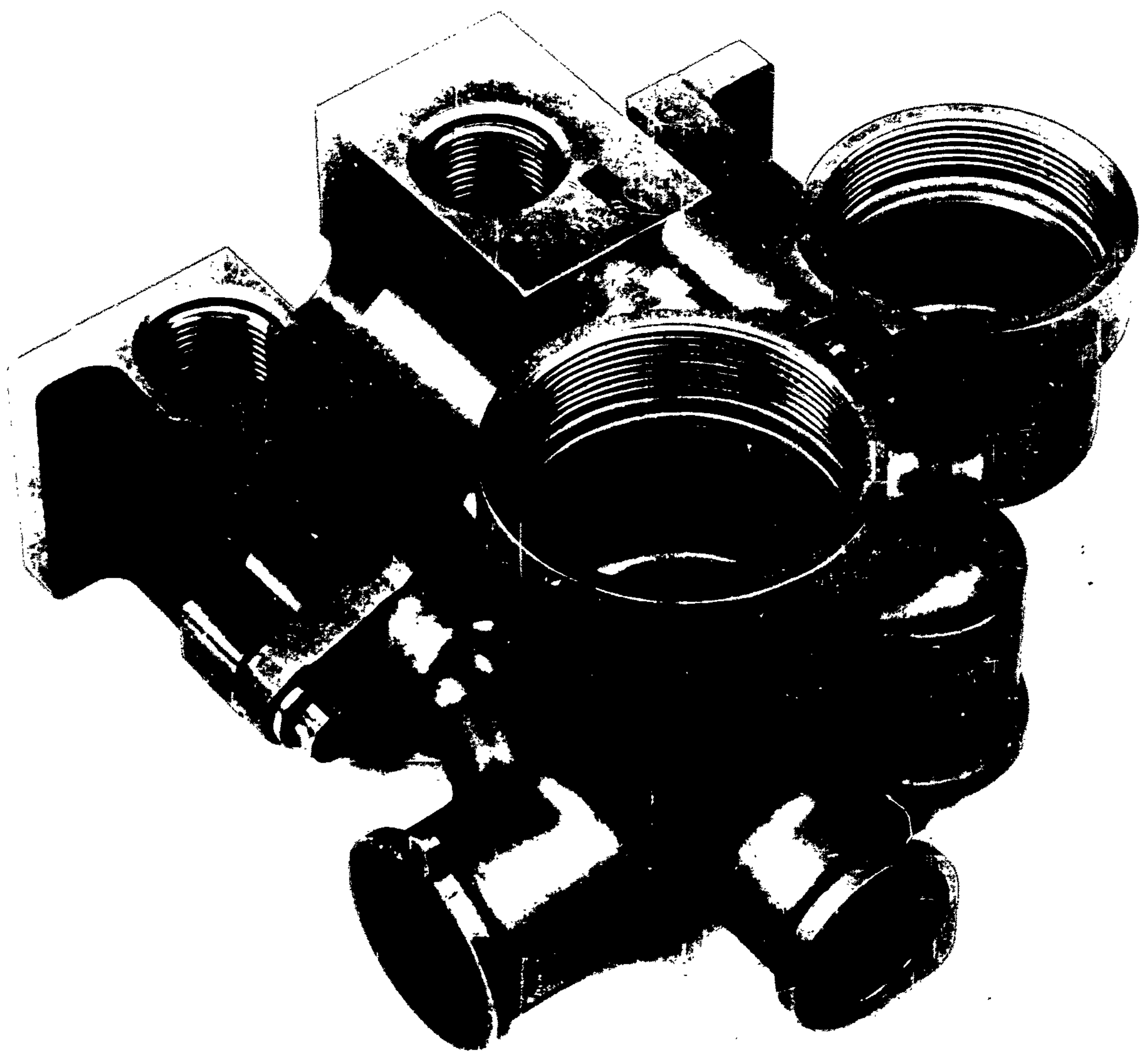


FIGURE III- 19

CVS-54943 AND CVS-54944 MODULAR HYDRAULIC MANIFOLDS FOR PACKAGE #5. FOR USE WITH CLASS B AND CLASS C FILTERS, SOLENOID - OPERATED SHUT-OFF VALVE, PRESSURE RELIEF VALVE AND PRESSURE SWITCH. LOWER RIGHT-HAND VIEW.

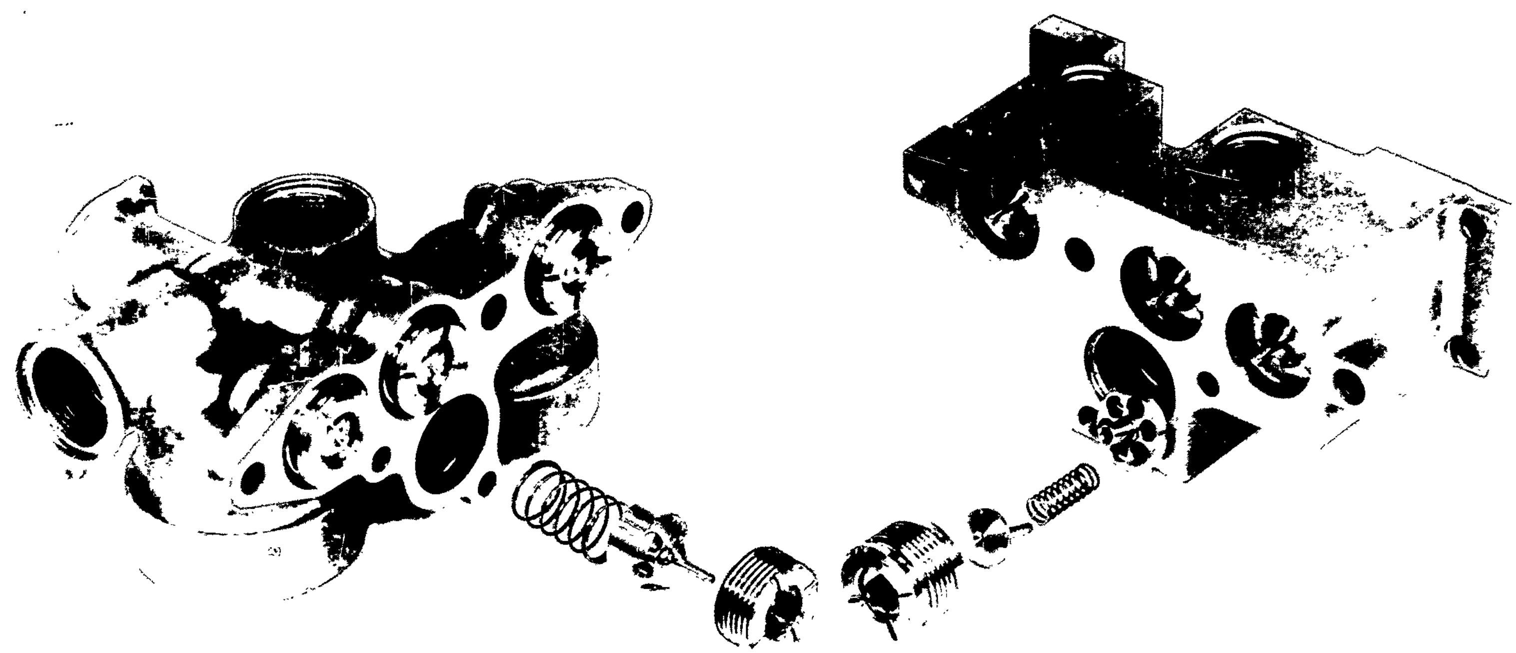


FIGURE III-20

CVS-54943 AND CVS-54944 MODULAR HYDRAULIC MANIFOLDS SEPARATED TO REVEAL AUTOMATIC SHUT-OFFS. THE MATING SHUT-OFFS FOR THE PRESSURE INLET PORT ARE SHOWN IN EXPLODED ARRANGEMENT.

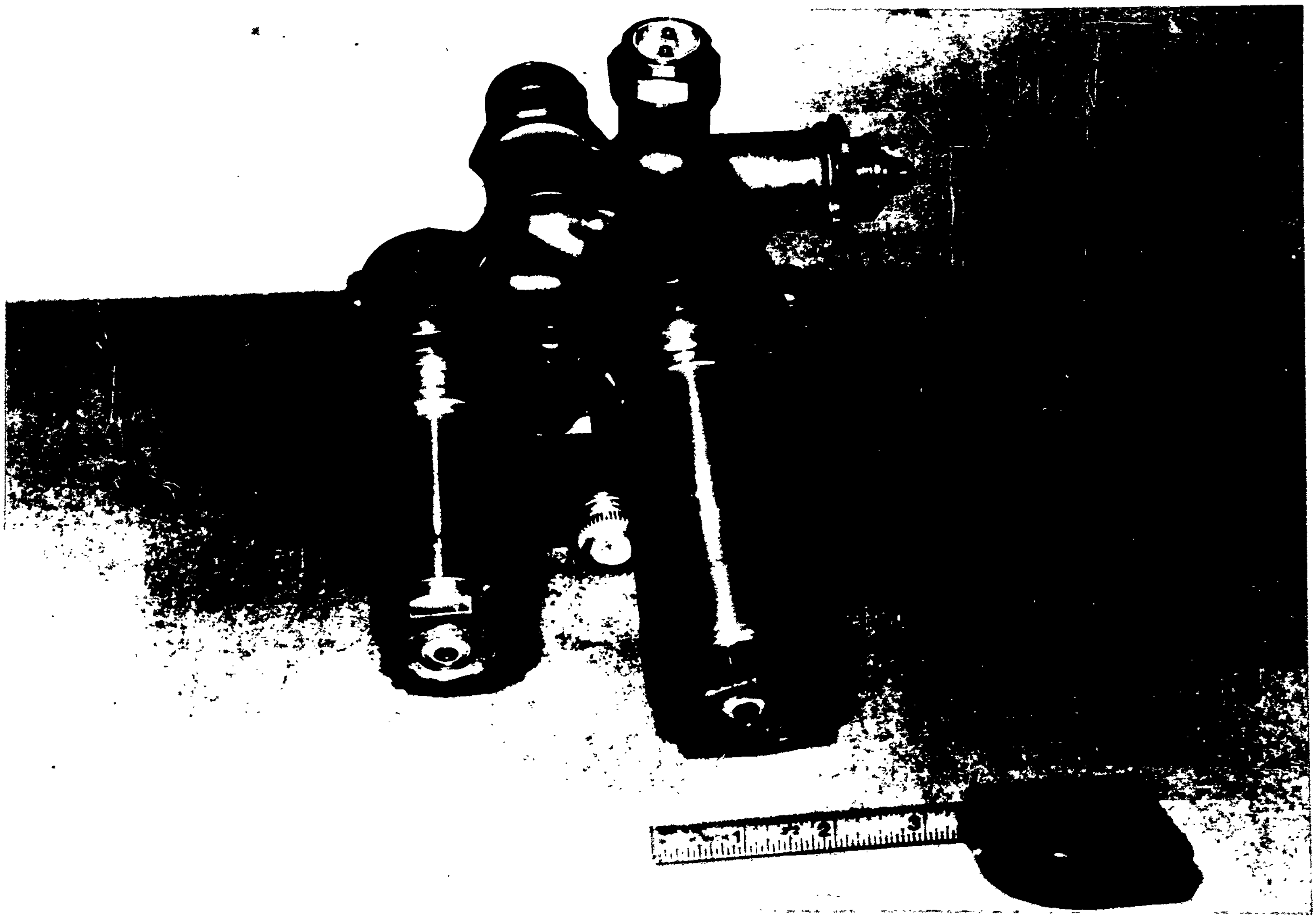


FIGURE III-21
#5 PACKAGE

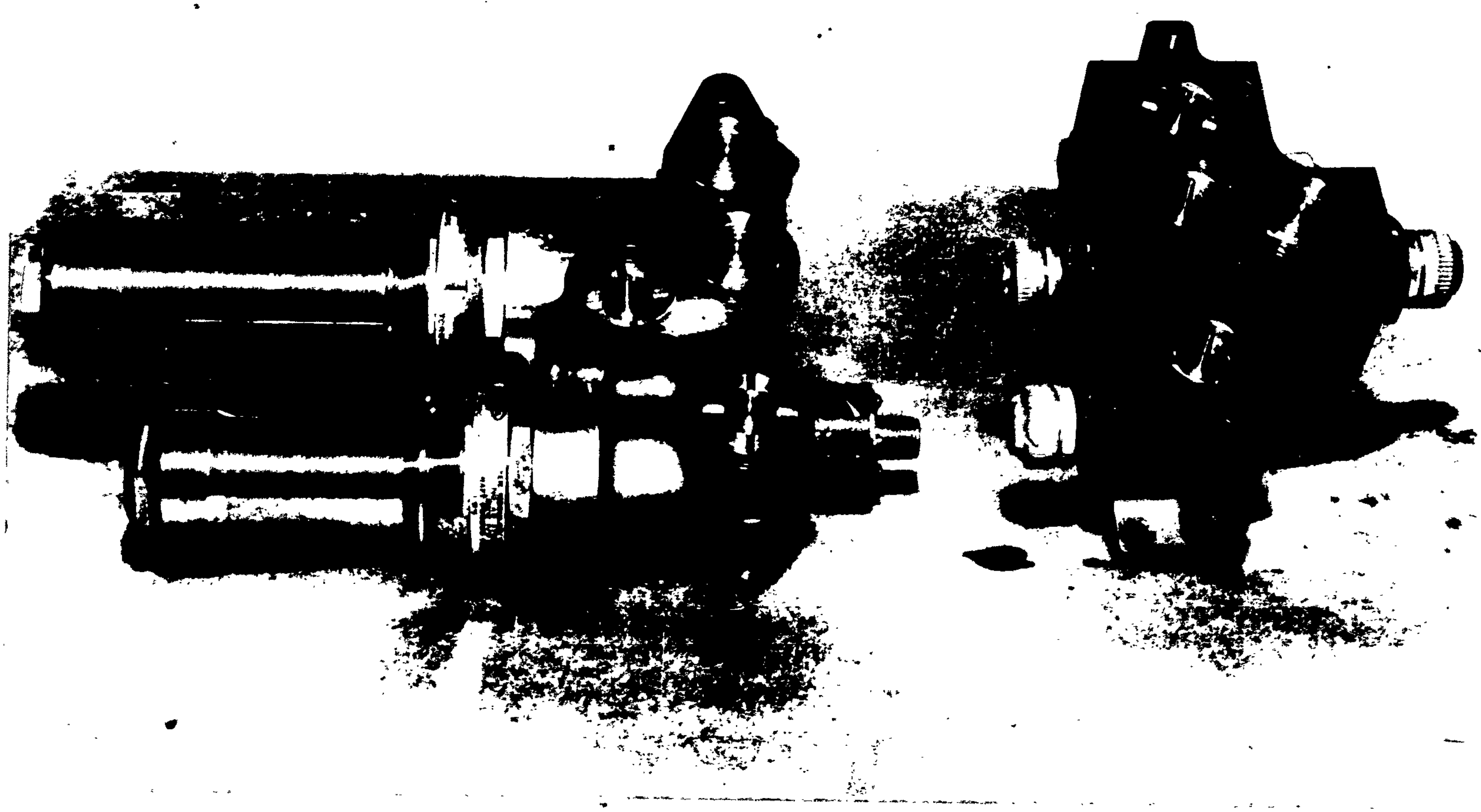
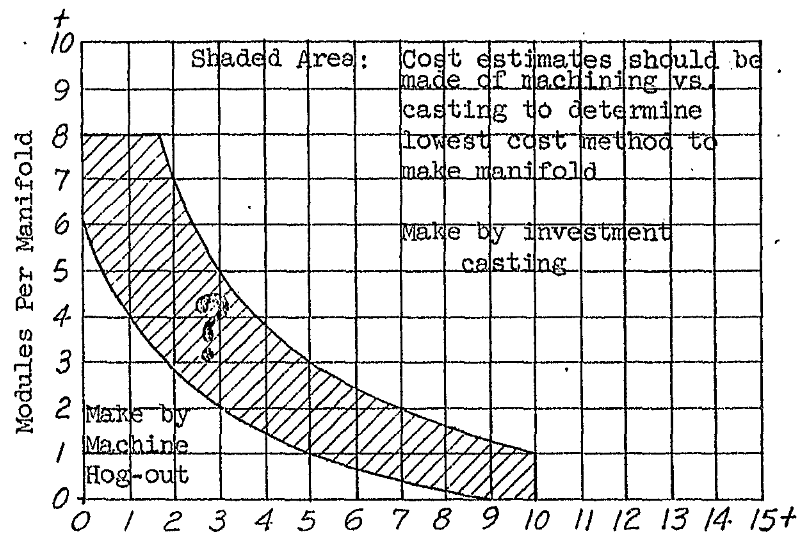


FIGURE III-22
#5 PACKAGE

Manifold Castings - At the outset of the modular hydraulic program, the exact configuration of the manifolds to be required for the test packages was not known. Casting, forging and hog-out configurations were considered as well as the type of material which would be most advantageous. Since that time, it has become evident that investment cast manifolds are decidedly superior to any other method of fabrication where a sizeable number of manifolds are involved. Figure III-24, below, is a guide which shows when it is profitable to either hog-out or cast manifolds.



CASTING VS. HOG-OUT COST COMPARISON
FIGURE III-24

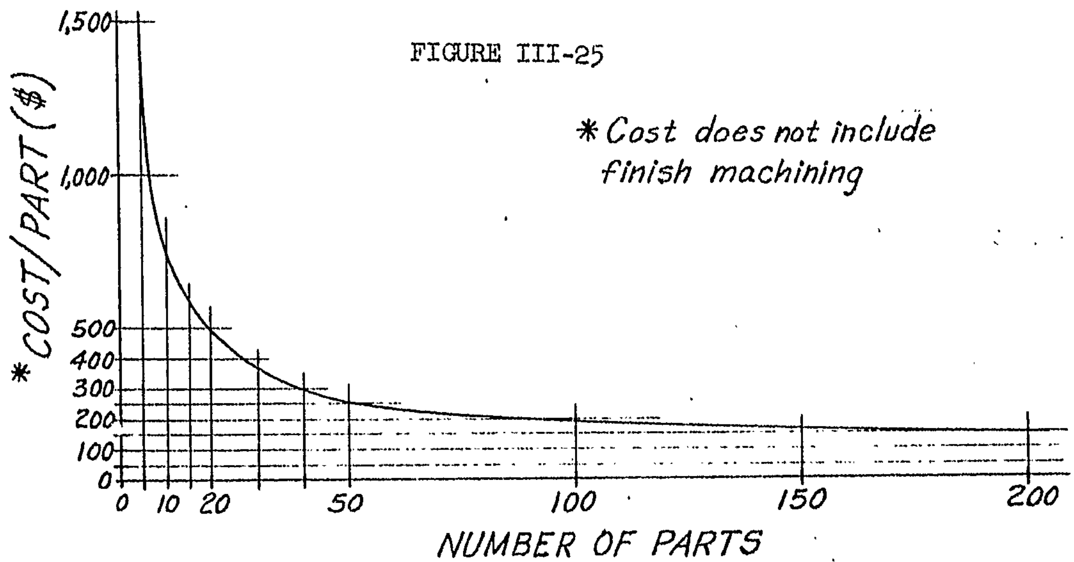
Production drawing ozalids of the #4 package manifold, CVS-54926-2, and the #5 package removable manifold, CVS-54943-2, were sent to several casting vendors for cost quotes. Average values are shown in Table III-1, below.

TABLE III-1

ITEM	COST ($)*	
	CVS-54926-2	CVS-54943-2
Tooling	5,808	5,815
5 parts	183 each	226 each
10 parts	166 each	152 each
50 parts	156 each	124 each
200 parts	133 each	107 each

* Cost does not include finish machining.

The curve shown in Figure III-25, below, is a curve of number of investment steel castings versus the cost per casting. This curve is based on the cost quotes from Table III-1.



A study of this curve shows that this particular casting would cost $1,366 each on an order of five parts. However, based on an order of two hundred parts, this casting would cost only $149 each. This represents more than a ninefold decrease in the cost of the part. The forged and hogged-out configurations cannot provide as great a decrease in cost per part. This fact, plus the inherent superiority of a cast part from the weight-saving standpoint, justifies the use of cast parts over forged or hogged-out parts whenever an appreciable number of parts are involved. Pertinent and conclusive data on casting techniques and materials best suited for Type III modular hydraulic manifolds is, at best, difficult to find and evaluate. Casting materials and techniques for a Type II system are readily available due to the aircraft industry's experience with non-ferrous castings which are acceptable for a Type II system. A Type III system, however, requires ferrous castings of a high quality which must be able to withstand 650°F and preferably maintain their strength at this temperature. The MLO-8200 hydraulic fluid used at this temperature, plus the temperature itself, prevents adequate protection of non-stainless steels from the elements. For these reasons, stainless steel materials are employed for manifolds of Type III systems. Research is being continued on available stainless steels suitable for casting and on the casting technique itself. Since the scope of this program does not embrace physical research into stainless steel investment casting, all data must necessarily come from existing reports and periodicals of physical research programs and from trade literature. Table III-2 on the following page lists some of the more common cast stainless steels presently cast by various vendors throughout the country.

Of the five manifolds designed for the modular hydraulic program, four are fabricated from 17-4PH stainless steel and one is fabricated from 6Al-4V titanium (MIL-T-9046C). Although none of the manifolds

TABLE III-2

MATERIAL			CASTABILITY	Maximum Operating Temp °F
Alloy Type	SAE Equivalent	AMS Equivalent		
17-4PH	-	5355	Good	1,500
302	60302	5358	Good	1,600
304	60304	5639*	Good	1,600
309 Modif.	60309	5650*	Excellent	Interm.
310	60310	5366	Excellent	2,000
316	60316	5360	Good	1,600
420	60420	5621*	Good	1,200
431	60442	5353	Good	1,500
440A	51440A*	5631*	Good	1,500
440C	51440C*	5352	Good	1,500

* Denotes specs for wrought forms rather than cast forms.

have actually been cast, all but one of the 17-4PH manifolds have been hogged-out to simulate typical investment-cast parts. The 17-4PH material was chosen for its high strength properties at elevated temperatures to afford the lightest possible design. While other stainless steels would permit harder sealing surfaces, better machining properties, etc., it was felt that the savings in weight gained by the use of 17-4PH would offset its disadvantages.

Permanent Plugs - Two types of permanent plugs were investigated for the use of sealing drilled passages in manifolds. The Lee plug proved to be more suitable than the Strataflex plug and was selected for use in the modular hydraulic program. It will be noted that package #2 required the use of one Lee plug and that package #4 required three Lee plugs. Use of this type plug in lieu of types similar to AN-806 or MS-21913 results in a significant space and weight savings.

The results of preliminary studies on the Lee plug and Strataflex plug are included in Appendix III-3. It was concluded that the Lee plug is a reliable means of plugging port holes under the temperature and pressure conditions of the modular program provided that:

(1) The port has a reasonable wall thickness (minimum of approximately 1/3 of plug diameter)

(2) Port tolerances are accurately controlled

(3) Port material is sufficiently hard.

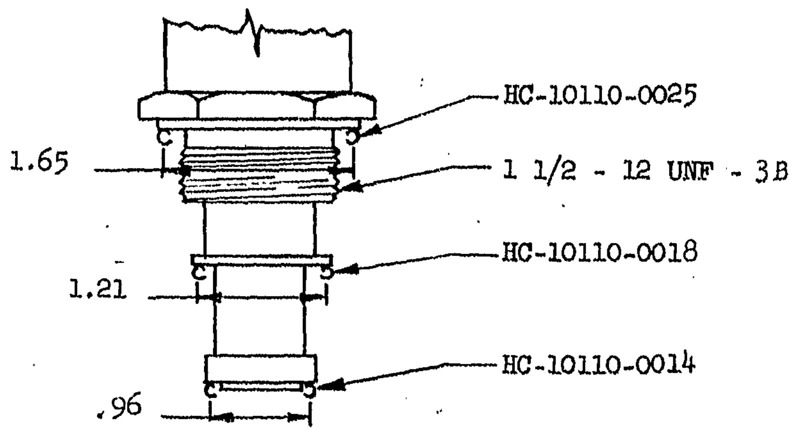
Component Locking Devices - Two friction locking devices, the Heli-Coil locking insert and the Hot-Loc, were studied as a means of locking the modular components in the manifolds. The results of these tests are included in Appendix III-4. Both of these devices showed much promise as a friction locking device. However, torque readings taken during metallic seal tests showed that a high torque would be required to install modular valves because of the force required to squeeze the seals, friction between the seals and sealing surfaces, and friction in the threads. This torque is more than equivalent to that which would be induced by any friction locking device. Because of this, it was decided not to use friction devices to secure modular valves in the manifolds. Experience in working with and qualifying the components has shown that the compression forces of the metallic seals and the friction forces mentioned above are more than adequate to lock the component in the manifold. Safety wire provisions were made on all four packages to secure all modular components.

Component Installation Torque - As previously mentioned, the torque required to install some of the components is rather high. Values that have been measured at Chance Vought are included in Table III-3 for design and reference purposes. A method for calculating torque to install components when using the 0.015 wall, 304 material Hi-Ceals is shown for information.

TABLE III-3

Modular Component	Class	Installation Torque Inch Pounds
Restrictor	A	468
	B	696
Check Valve	A	486
	B	844
	C	945
Shuttle Valve	A	1300
	B	-
	C	3000
Pressure Switch	A	550
Thermal Relief	A	460
	B	360
	C	840
Priority	B	2030
	C	1600
Pressure Relief	A	1080
	C	1600
Filter	A	2320
	B	2560
	C	2960
Sequence	A	-
	B	1560
3-way, 2-position	A	1880
2-position Shut-off	A	1390

A method used to calculate the installation torque when using Hi-Ceals, which proved to be fairly accurate, is shown below.



3 Way - 2 Position Selector (Class A)

1. Radii to points of seal contact: 0.82, 0.60, 0.48

2. From compression tests, force to deflect Hi-Ceal 0.014 inch (assumed average deflection) is:

$$300 \frac{\text{Pounds}}{\text{Inch of Circumference}}$$

3. Required force:

$$
\begin{aligned}
-0025 &= \pi\,(1.65 \text{ in.})\ 300\ \#/\text{in.} = 1550\# \\
-0018 &= \pi\,(1.21 \text{ in.})\ 300\ \#/\text{in.} = 1140\# \\
-0014 &= \pi\,(.096 \text{ in.})\ 300\ \#/\text{in.} = \underline{910\#} \\
&\qquad\qquad\qquad \text{Total} = 3600\#
\end{aligned}
$$

4. Torque caused by Hi-Ceal friction:

$T = \mu P r$

T = torque in #
μ = coefficient friction
P = load
r = radius

$$
\begin{aligned}
-0025 &= T = \mu Pr = .4\ (1550)\ .82 = 505 \text{ in. } \# \\
-0018 &= T = \mu Pr = .4\ (1140)\ .60 = 274 \ '' \\
-0014 &= T = \mu Pr = .4\ (\ 910)\ .48 = \underline{174}\ '' \\
&\qquad\qquad\qquad \text{Total} = 856 \text{ in. } \#
\end{aligned}
$$

5. Torque caused by thread friction:

$$T_t = \delta_t W \frac{\cos\theta \tan a + \mu_t}{\cos\theta - \mu_t \tan a}$$

T_t = thread torque

δ_t = thread pitch radius

W = total load

θ = thread half angle (30°)

a = thread helix angle or $\tan a = \dfrac{\text{lead}}{\text{thread circumference}} = \dfrac{.0833}{\pi(1.46)}$

μ_t = coefficient of friction

Assume μ_t = .4

$$T_t = .72\ (3600\#) \left[\frac{(.866)(.0182) + .4}{(.866) - (.4)(.0182)} \right]$$

$$T_t = 2590 \left(\frac{.4157}{.866 - .0073} \right) = 2590\ (.480)$$

$$T_t = 1240 \text{ in. } \#$$

TOTAL TORQUE = 1240 + 850 = 2090 in. #

This compares to a measured value of 1880 in. #. A similar calculation for the Class C shuttle indicates required torque to be 2940 in. # compared to 3,000 measured.

Thread Lubricants - From the preceding calculations it is seen that a large part of the torque required to install modular components is utilized to overcome thread friction. For example, if the coefficient of thread friction is reduced from 0.4 to 0.2, the thread torque is reduced from 1240 inch pounds to 640 inch pounds; and, the total component installation torque is reduced from 2090 inch pounds to 1490 inch pounds. Besides reducing component installation torque, the lubricant is valuable in eliminating thread galling caused by high loads required to deflect the metallic seals.

Several lubricants were tested in conjunction with a 2-1/4 - 12 UN male plug and test housing in an effort to determine the best lubricant to use for module installation. The housing material was CRES 17-4PH with a hardness of Rc 42, and the plug material was CRES 321 with a hardness of Rc 50. In general, the lubricant must be compatible with MLO-8200 hydraulic fluid, must not introduce contamination to the system, and must provide a low coefficient of friction. The plug is shown in Figure III-26. A diagram of the test setup and curves of break-out torque versus axial thread load are shown on Figure III-27 . The test procedure was to (1) insert the plug in the manifold until seven threads were engaged; (2) connect the loading

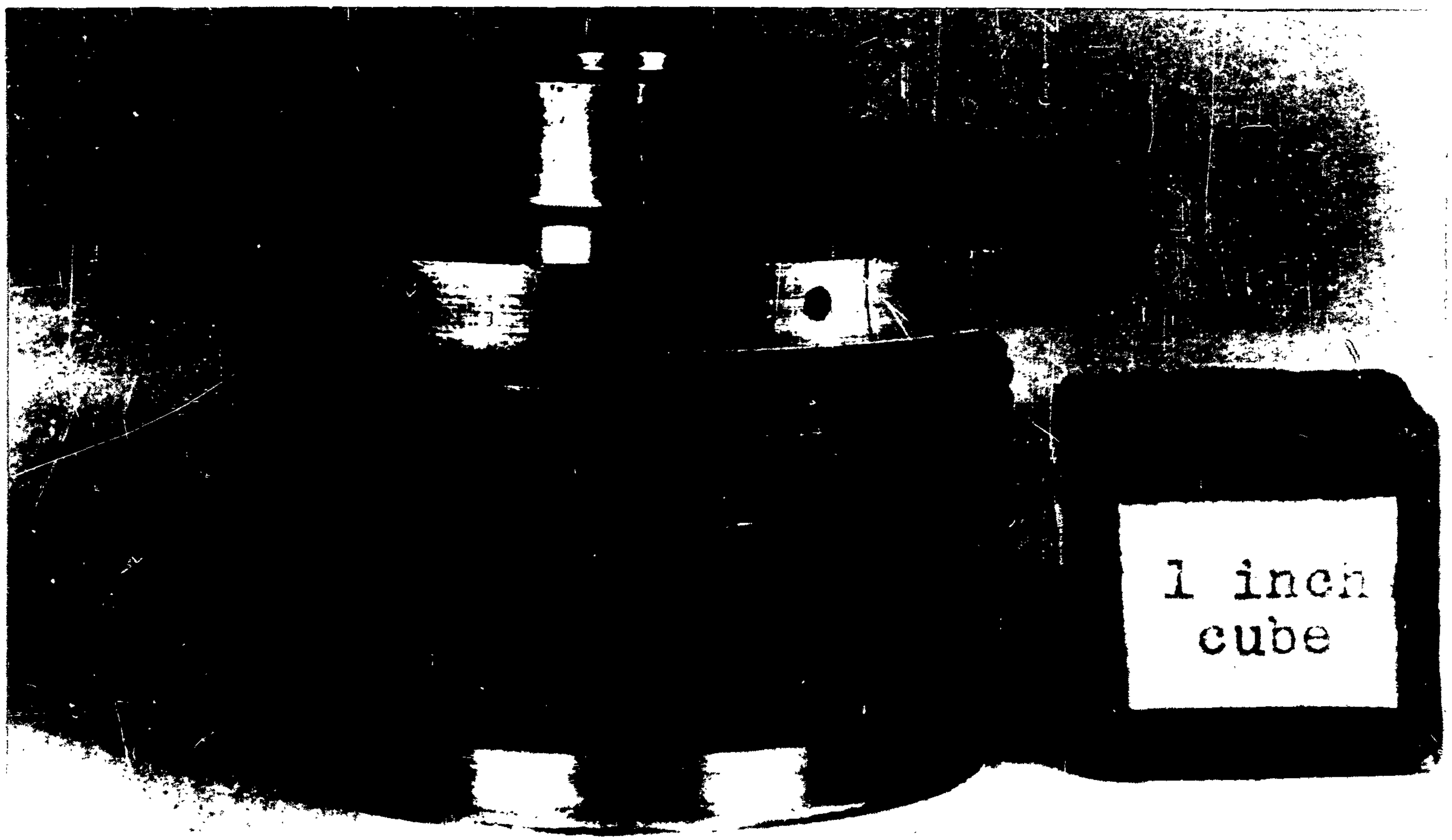


FIGURE III - 26

MODULAR HYDRAULIC TEST CAP WITH TEFLON ENAMEL COATED 2-1/4-12UNF THREADS, AFTER COMPLETION OF 4 THREAD FRICTION TESTS.

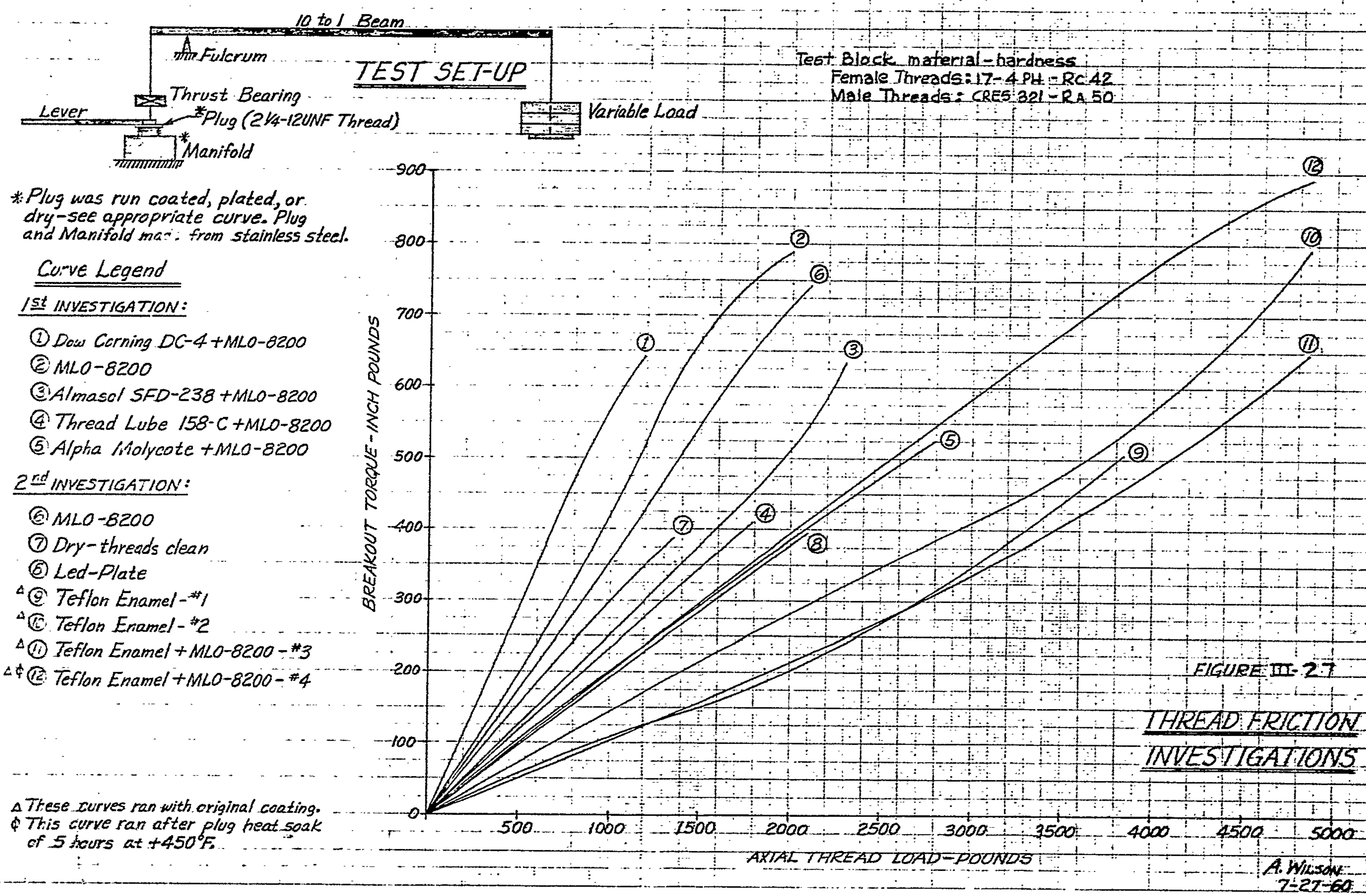


FIGURE III-27

THREAD FRICTION INVESTIGATIONS

mechanism to the plug; (3) vary the load on the beam and measure the torque required to increase the thread engagement.

Lubricants tested were as follows:

1. DC-4 manufactured by Dow Corning Corporation, Midland, Michigan -- not adequate; showed tendency to allow thread galling.

2. Bonded Alpha Molycote, MoS_2, manufactured by Alpha Molycote Corporation of Stanford, Connecticut -- good lubricant, but introduced particle contamination.

3. Thread Compound 158-C manufactured by Lubrication Engineers, Incorporated, of Ft. Worth, Texas -- good lubricant but introduced particle contamination.

4. Almasol SFD-238 manufactured by Almasol Corporation, a division of Lubrication Engineers, Inc., of Ft. Worth, Texas -- excellent lubricant but introduced particle contamination. Analysis of the contaminant, however, indicated 98% to be less than 1 micron in size.

5. Lead-Plate Compound manufactured by Armite Laboratories of Los Angeles, California -- Lead-Plate is basically fine lead particles in a special grease suspension. The lead particles are of a size to contaminate the hydraulic system, however it is felt that the manufacturer may be able to provide a modified "Lead-Plate" which would be acceptable. This compound is advertised to be effective up to 3,000°F, which may certainly have future applications. A reduced particle size "Lead Plate" was not investigated.

6. One-Coat Teflon Enamel manufactured by E. I. duPont de Nemours of Wilmington, Delaware -- Classified as a spray enamel. It is green in color and as applied per specification CVC 67 has a film thickness of 0.0007 ± 0.0001. The teflon coating is duPont's TFE fluorocarbon resin and has the typical soapy feel of solid teflon. It does not chip or flake under abrasion, has a marked resistance to peeling, and no tendency to "transfer" to a mating part even when subjected to high bearing pressures.

"Lead-Plate" and "One-Coat Teflon Enamel" were more effective as anti-friction and anti-galling agents than any plating or coating tested, with the Teflon Enamel being superior to "Lead-Plate." The "Lead-Plate" was tested by coating the plug threads with the compound and running the plug on a dry manifold. The Teflon Enamel was tested by coating the plug threads with the enamel then testing as follows:

Trial #1 - Teflon-coated plug running on dry manifold up to 3,840 # axial load.

Trial #2 - Teflon-coated plug running on dry manifold up to 4,850 # axial load.

Trial #3 - Teflon-coated plug running on manifold with MLO-8200 hydraulic fluid up to 4,850 # axial load.

Teflon-coated plug and manifold were next preloaded to axial load of 4,850 #, soaked in MLO-8200 and held at 450°F for 5 hours.

Trial #4 - Teflon-coated plug running on manifold with MLO-8200 residue up to 4,850 # axial load.

Although the teflon coat held up very well during repeated tests, some peeling did occur. It is concluded that of those tested, teflon coat is the most satisfactory; however, all would contribute contamination to the hydraulic system. The development and qualification tests of all the modular components were completed without the use of a thread lubricant. The one exception was the 4-way, 3-position, cartridge-type selector valve which has external threads (i.e., the threads of all the other components are within the hydraulic system when installed). To prevent thread galling and to reduce installation torque, the following method was recommended and successfully used:

(1) Install the modular valve with only one metallic seal installed and apply torque until the valve bottoms out.

(2) Remove the valve and the seal and install the second seal, presetting it in the same manner as the first.

(3) Repeat this presetting process for a third seal.

(4) Install all of the preset metallic seals in their proper cavities and apply torque until the valve bottoms out.

The above method significantly reduces final installation torque and galling tendencies.

General Specification for Packaging Modular Components

Based on the experience gained in working with metallic seals, modular components, manifold and package designs, and package tests, a MIL type specification was prepared covering the general requirements for packaging modular hydraulic components. As previously mentioned, this specification is included in Appendix III-1.

Conclusions

It is concluded that reliable means of packaging modular components in 4,000 psi, 450°F hydraulic systems has been developed. However, the studies

made during this phase of the modular program point out many problems that the designer must be aware of. Before starting the design of any package, the designer should thoroughly familiarize himself with the "General Specification for Packaging Modular Components." Some of the problems that the designer may expect are:

(1) When designing a minimum weight manifold, the manifold will probably take such a complex shape that a casting will be the most economical approach for production quantities. When a manifold design is simplified to allow economical machining, significant weight increase results. Cost versus weight saving is a constant consideration in the design of manifolds.

(2) It must be noted that complete testing of a modular hydraulic package, which may have as many as five or more modules, is not as easily accomplished as individual component testing of conventional hydraulic systems. Once installed in a common manifold, modules will generally affect one another's function. Therefore, to isolate the performance of each module or to test for a certain module function, it will sometimes become necessary to employ test plugs in place of modules to accomplish the testing of a particular module. Due to the inter-related arrangement of various modules in one common manifold, considerable thought must be given to the testing procedure for the complete package in order to minimize the complexity of the package test setup.

(3) Two of the Chance Vought package designs #2 and #5, experienced leakage problems between faces of two manifolds that were bolted together. In each case, high loads were present which tended to spread these faces apart. Whereas an elastomer seal would tolerate this movement, the metallic seal would not. In designing for metallic seal face mount applications, the designer should strive for zero deflection. Deflections not only cause leakage problems, but fatigue failures of the metallic seal when they are subject to pressure impulses. In general, a screw-in design will give better results than a bolted-face-mount design when high loads are present.

(4) The pressure drop through the Chance Vought package designs was somewhat higher than originally anticipated. The higher pressure drop is mostly due to the complex flow path through the package. The designer must make larger flow passages and smooth out flow path if pressure drop must be small.

APPENDIX II-1

CVC M.H.2

GENERAL PROCUREMENT INFORMATION FOR METALLIC SEALS

CHANCE VOUGHT AIRCRAFT
INCORPORATED · DALLAS, TEXAS

ENGINEERING DEPARTMENT SPECIFICATION

ORIGIN. BY J. M. Spell	[signature] ORIGIN. GP. APP.	[signature] PROJECT ENGINEER		CVA M. H. 2
GROUP UNIT 54232	DATE 2-25	DATE 2-25-59	DATE	
REL. DATE ______ REL. DESK ______	DATE	DATE	DATE	ECP PC

GENERAL PROCUREMENT INFORMATION

FOR

STATIC METALLIC SEALS

1. GENERAL.- Chance Vought Aircraft, Incorporated, is interested in obtaining metallic seals for static applications in a 4000 psi hydraulic system. This brief brochure outlines some tentative design, manufacturing and testing considerations relative to the types of seal desired.

2. DESIGN AND MANUFACTURING REQUIREMENTS

2.1 Fluid.- The hydraulic fluid used shall be MLO-8200 which may be purchased from the Oronite Chemical Company, 200 Bush Street, San Francisco, California.

2.2 Temperature Range.- The seal shall be capable of full performance with the fluid at any temperature throughout the range of -65°F to 450°F and shall seal satisfactorily within a range of ambient temperatures from -65°F to 650°F.

2.3 Altitude.- The altitude pressure range shall be sea level to 100,000 feet.

2.4 Rated Pressure.- The rated hydraulic pressure shall be 4000 psi.

2.5 Re-usability versus Cost.- It is desirable that seals be of such a design as to permit their re-use a minimum of 15 times. during the life of the seal. If the seal is crushed or deformed by the installation process so as to require that it be discarded after one use, then its unit cost should be low enough to justify storage and replacement costs in service.

2.6 Machining Tolerances and Surface Finishes.- It is desirable that machining tolerances for hydraulic component sealing surfaces or cavity dimensions not be closer than ± .005 and that surface finishes finer than 32 RHA not be required for satisfactory sealing.

2.7 Special Tools.- The seal design should be such that special or unusual tools will not be required for normal installation and removal of the seal.

2.8 Lubricants.- No lubricant should be required for seal installation or performance other than hydraulic fluid MLO-8200.

2.9 Seal Tightening Forces.- It is desirable that a reasonable range exist between minimum and maximum tightening values for seal installation. It would be desirable, further, that the torque values for effective seal tightening using various thread sizes should not exceed the torque values for corresponding MS flareless fitting thread sizes.

3. TEST REQUIREMENTS

3.1 Proof Pressure.- Each seal shall be subjected to proof pressure of 6000 psi at 450°F for a period of 5 minutes. There shall be no leakage and no permanent deformation or other visible damage to the seal.

3.2 Burst Pressure.- The seal shall withstand a burst pressure of 10,000 psi at room temperature for a period of 5 minutes with no leakage.

3.3 Impulse Tests.- Th impulse tests shall be conducted at a rate of 35 ± 5 cpm. Each impulse cycle shall constitute a rise from 0 psi to surge pressure and drop to zero as shown in Figure I. Hydraulic fluid shall be used as a test medium with a peak surge pressure of 1.43 to 1.57 times the rated pressure as shown by an oscilliscope. The seals shall be tested for a total of 200,000 cycles per spectrum in Figure II. Before and after the impulse tests the seals should be proof pressure tested at 450°F for 5 minutes. There shall be no leakage during the impulse tests or proof pressure tests.

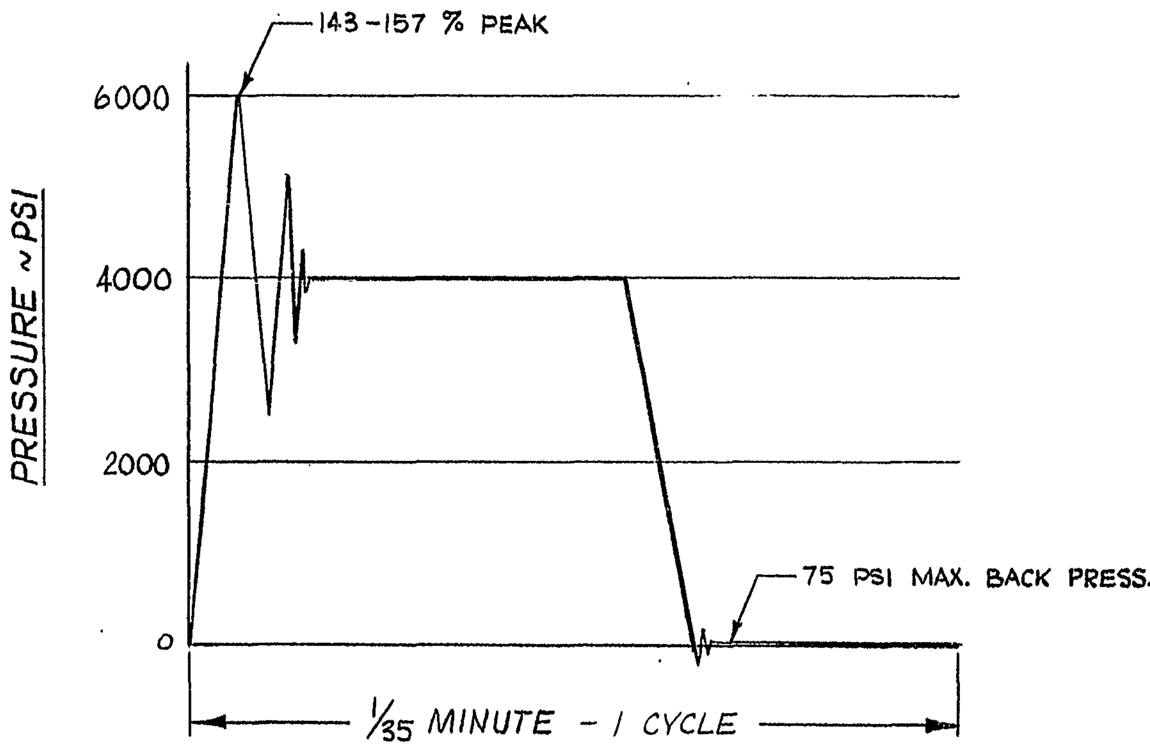
4. SEAL TYPES AND CAVITY CONFIGURATIONS

4.1 General.- Seals of the types mentioned below are desired in sizes from 3/8 inch O.D. up to and including 3 inch O.D. in increments of 1/8 inch. It is further desired that the seals be of double-acting design, that is, capable of withstanding pressure applied from either direction against the seal.

4.2 Boss, Step, or Face Seal.- Chance Vought Aircraft is interested in testing and evaluating existing metallic seals of a boss, step, or face type. Seals of this type may use AND 10050 port dimensions although this is not a requirement. Any other type of seal cavity may be used depending upon the requirements of the seal design.

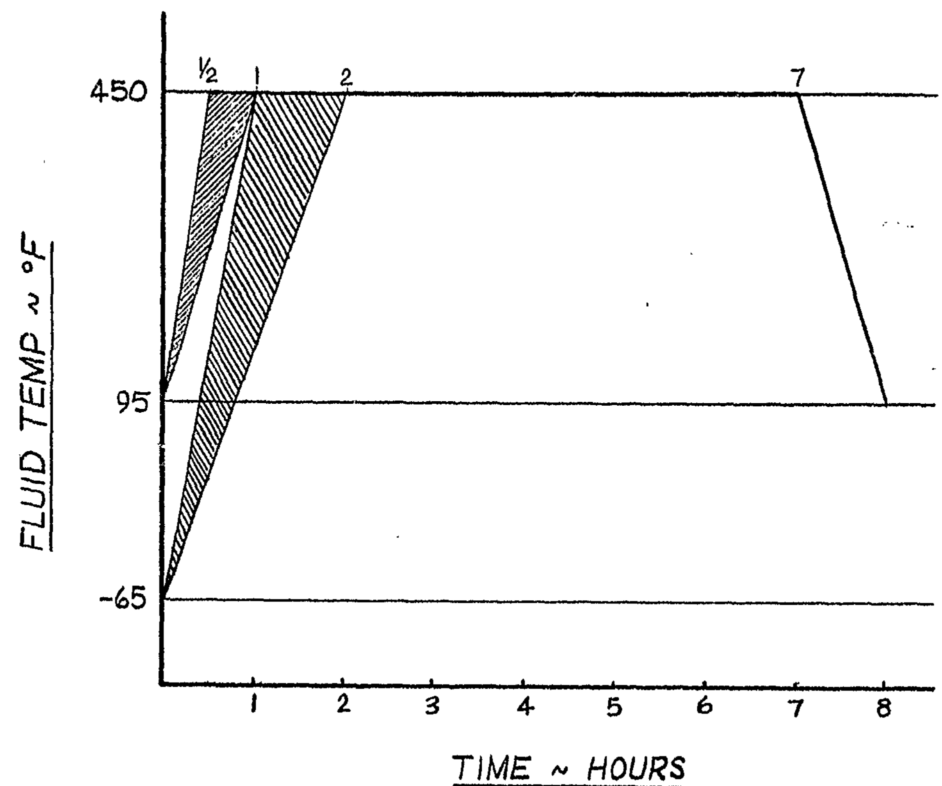
4.3 Radial, Circumferential, or Annular Seal.- Chance Vought Aircraft is interested in obtaining proposals for the development of seals of this type. The attached sketches mentioned below show the sealing applications for the type of seal desired.

4.4 Seal Cavity Configurations.- Attached are sketches representing the types of seals outlined above and showing, in addition, the general configuration of the hydraulic components with which the metallic seals are to be used.



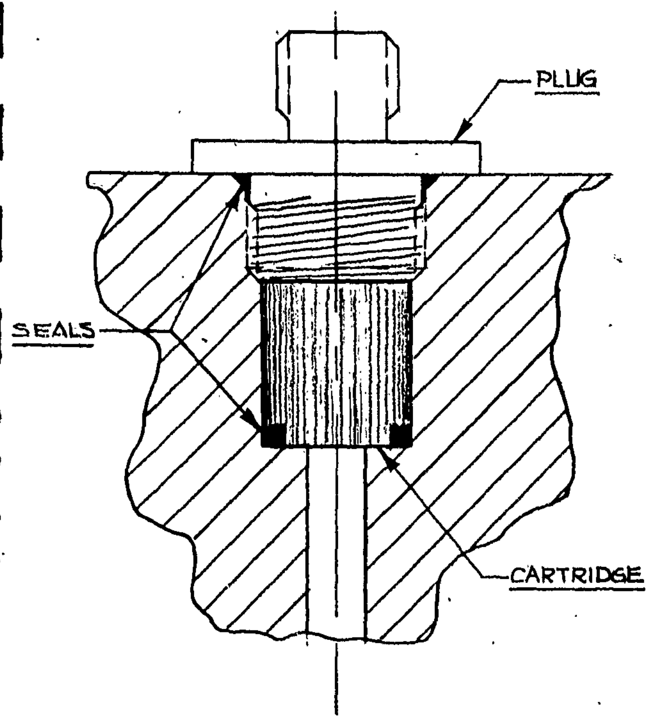
Approximate pressure-time cycle for
impulse testing static metallic seals.

FIGURE I

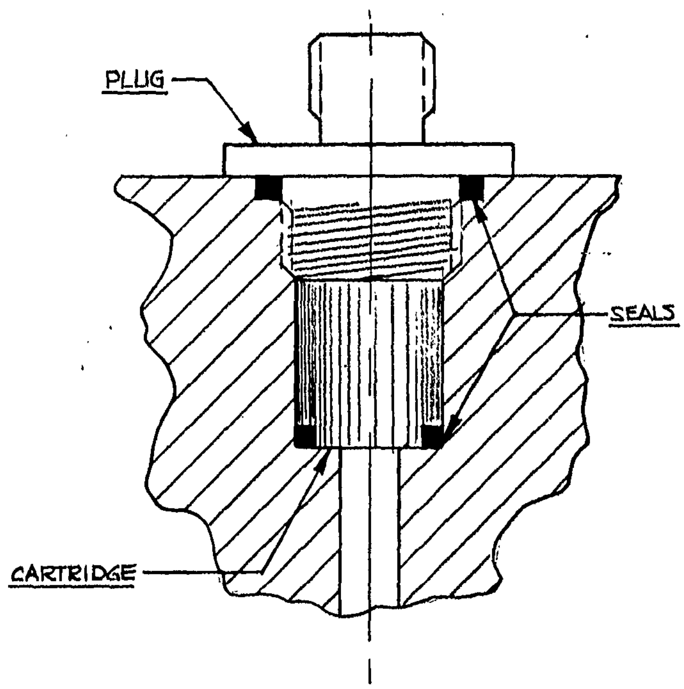


The impulse test (200,000 cycles) shall be conducted while the seal undergoes the temperature-time spectrum shown above. Each spectrum should take approximately eight hours to complete and should consist of approximately 16,670 cycles. Twelve days are required for the test. The first, fourth, seventh and tenth spectrums shall begin after the test set-up has soaked at -65°F for 8 hours (overnight). The remaining spectrums shall begin at 95° ± 25°F on the 2nd, 3rd, 5th, 6th, 8th, 9th, 11th and 12th days of the test. The rate of temperature rise shall be within the shaded areas shown on the curves above.

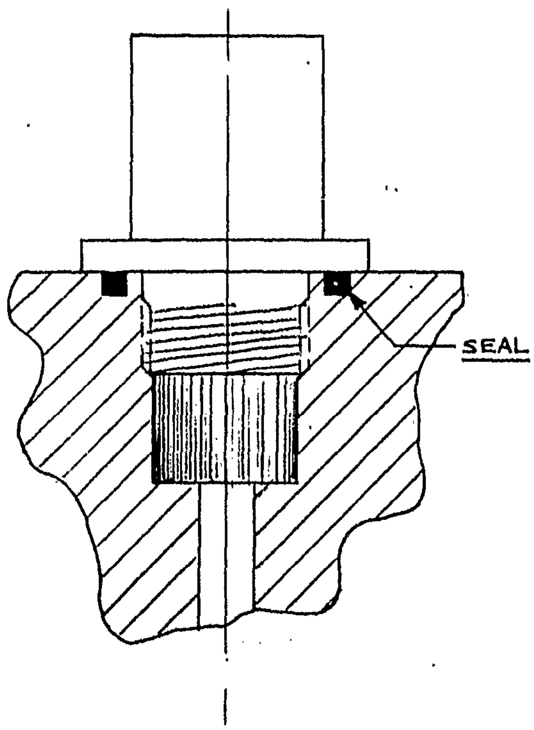
FIGURE II



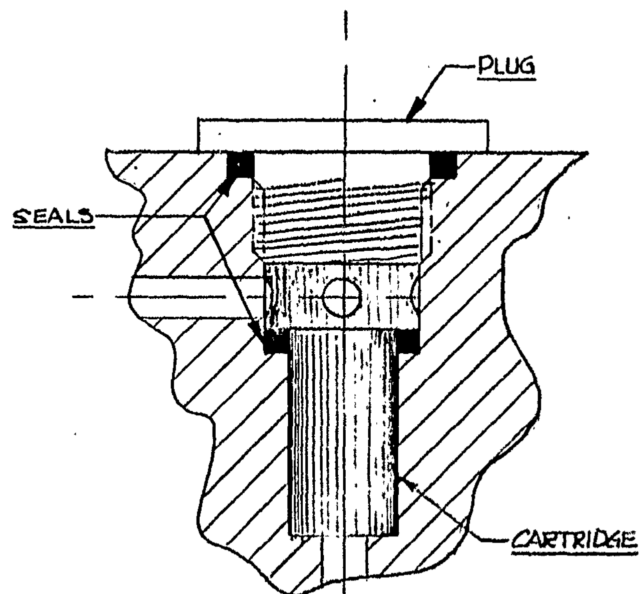
COMBINATION BOSS AND STEP SEALS



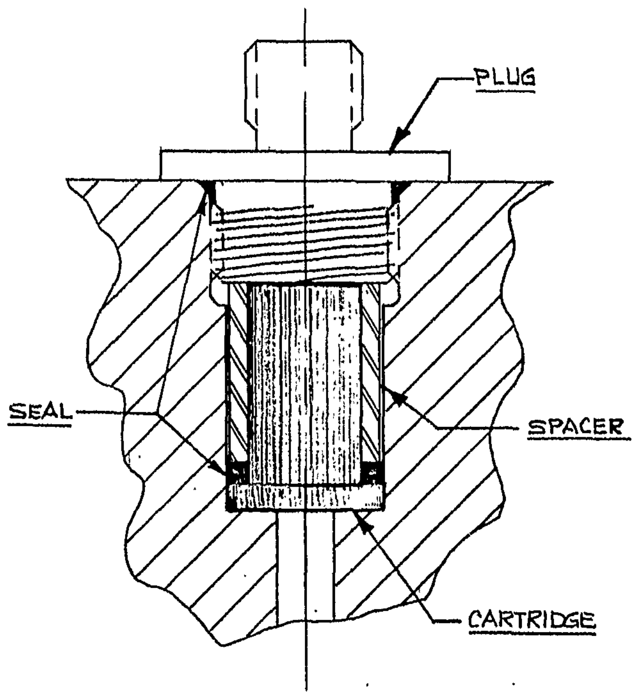
STEP SEALS



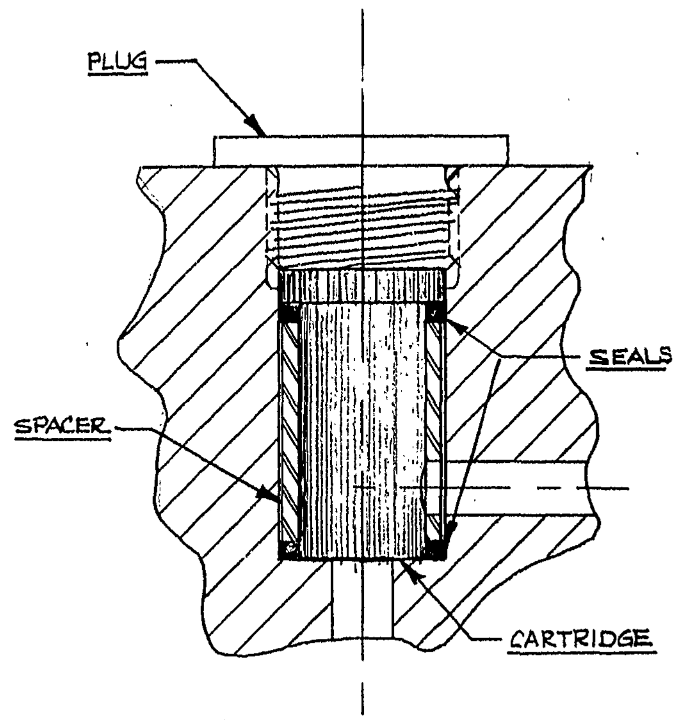
FACE SEAL



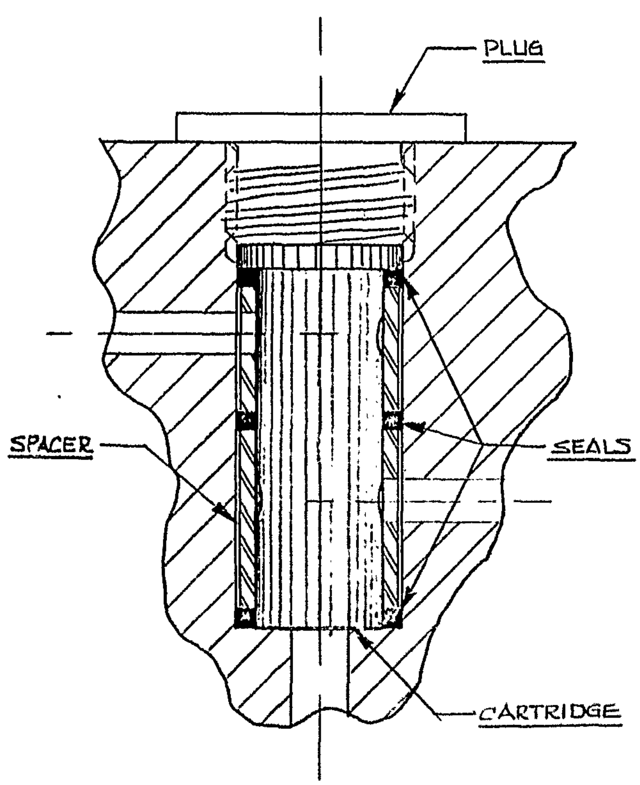
DOUBLE - STEP SEAL



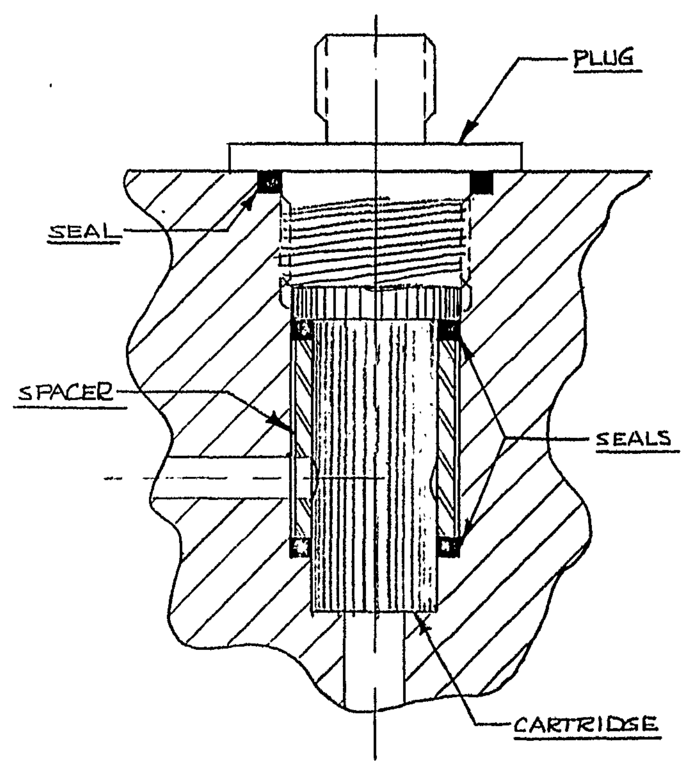
COMBINATION BOSS AND RADIAL



DOUBLE - RADIAL



TRIPLE - RADIAL



COMBINATION STEP AND RADIAL

APPENDIX II-2

SEAL MANUFACTURERS CONTACTED

ORIGINAL 20 SEAL VENDORS CONTACTED (* denotes reply received)

* 1. Advance Products Company
* 2. Aero Gasket Corporation
3. Anagra, Inc.
4. Arnav Aircraft Associates, Inc.
* 5. Auburn Manufacturing Company
* 6. Bendix-Pacific Division Bendix Aviation Corporation
* 7. Double Seal Ring Company
* 8. D.S.D. Manufacturing Company
* 9. Fulton Sylphon Division Robert Shaw-Fulton Controls Co.
10. Gasket Engineering Company, Inc.
11. Garlock Packing Company
* 12. Harrison Manufacturing Co.
13. Johns-Manville Corporation
* 14. Joseph Soraghan & Associates
15. Koppers Company, Inc.
16. Linear, Inc.
* 17. Muskegon Piston Ring Company
* 18. Navan Products, Inc.
* 19. On Mark Couplings
* 20. United Aircraft Products, Inc.

ADDITIONAL VENDORS CONTACTED (* denotes reply received)

* 1. Adel Precision Products
* 2. Aeroquip
* 3. Aircraft Porous Media
4. American Hammered
* 5. Cadillac Gage Company
* 6. Guy Whitaker Company, Inc.
* 7. Hydrodyne Corporation
8. Perfect Seal Manufacturing Company
9. Sealed Power
* 10. Vinson Manufacturing Company

APPENDIX II-3

GENERAL TEST PROCEDURE FOR METALLIC SEAL

GENERAL TEST PROCEDURE
FOR METALLIC SEALS

Ref: (a) CVA MH-2 dated 25 February 1959

PURPOSE

The purpose of this outline is to provide a working test procedure to evaluate metallic seals suitable for use in a hydraulic system using MLO-8200 hydraulic fluid in a temperature range of -65°F to 450°F and an operating pressure of 4000 psi.

SCOPE

Samples of all boss and face type metallic seals that appear to meet the requirements of reference (a) shall be procured for evaluation. Boss type seals shall be evaluated in the -6, -10, -16, -20, and -28 sizes in a test manifold shown in the photograph on page 12. Face type seals having inside diameters approximately equal to inside diameters of AN 6227-8, -12, -19, -23, and -30 shall be evaluated. The test manifold for each seal design is shown in TL-3618, page 11. Where reference is made to torque, it shall be understood as having the same meaning as squeeze for face seals when applicable. Normal pressure application is defined as meaning the application of a differential pressure across the seal with the pressure being greatest on the inside diameter. For reversed pressure, the pressure shall be greatest on the outside diameter of the seal. Seal testing shall be performed in two parts as outlined. Only those seals successfully completing Part I tests will be tested in accordance with Part II outline. Leakage across internal seals shall be channeled to prefilled clear tubing stand pipes. Fluid level will be marked before cycling is commenced each day and any fluid level change will be noted.

I. Preliminary Tests

A. Physical Check

1. All seals and fittings used shall be assigned a serial number before testing begins. Serial numbers shall start with 1 and run consecutively for each type and size seal or fitting used. Test fittings shall be stencilled with serial numbers and seals shall be identified by attaching metal serial number tags.

2. Each test seal and fitting used with a test seal shall be visually examined for any unusual defects or deviations from blueprint requirements. A record shall be kept on all visual observations.

3. All critical dimensions affecting sealing properties shall be measured on seals and fittings. The measured dimensions will be recorded for comparing with design requirements.

B. Leakage and Proof Pressure

1. Install test seals in appropriate manifolds or fittings according to the vendor's recommended installation procedure. Where a range of torque values is specified, the lower torque value shall be used first. Maintain a record of torque applied on each test seal.

2. Apply static hydraulic pressures of 10 psi, 4000 psi, and 6000 psi on the seals. Maintain each pressure for a period of at least 10 minutes and note any leakage which may occur.

3. If leakage occurs while the seal is under static pressure, the pressure shall be removed and the seal torque increased by approximately 10% of the original torque value, or some reasonable squeeze value in case of face seals, when higher torque values are conducive to better sealing. The higher torque values shall not be applied before a seal has undergone leakage checks at all three pressure levels, unless excessive leakage deems this to be impractical. In general, when leakage occurs, a higher torque or squeeze shall be applied to the seal until leakage ceases except that the torque applied should not exceed 400% of the maximum torque recommended by the vendor.

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4. If leakage has not been stopped after reaching the 400% of maximum original torque value, the seal shall be removed from the set-up and all mating surfaces examined for any possible defects that might cause leakage.

5. If cause for leakage is found to be in the mating fitting to the seal, then the cause shall be remedied and leakage testing of seal started over provided seal has not been damaged. If no cause for leakage is evident, then the seal shall be reassembled on the same fittings and leakage testing of seal started over. On the second assembly, special care must be taken to assure that seal is properly seated. If seal continues to leak, no further testing will be performed.

6. For those seals considered suitable for withstanding reversed pressure, the test procedure of Paragraph I-B-1 through I-B-5 shall be repeated with pressure from the opposite direction.

C. Impulse Tests

1. Seals satisfactorily completing leakage tests shall undergo 50,000 pressure impulse cycles while the seals undergo the temperature-time spectrum shown in Figure I, page 8. The test shall be composed of four complete temperature-time spectrums. The pressure impulse cycling shall conform as nearly as possible to that shown in Figure II, page 9. The pressure impulse set-up will be as shown schematically in Figure III, page 10. Pressure impulsing during the first and third temperature-time spectrum runs shall start when the seal temperature is between 70°F and 100°F. The pressure impulsing during the second and fourth temperature-time spectrum shall start when the seal temperature is at -65°F. For those seals considered suitable for withstanding reversed pressure, impulse pressure application shall be in the opposite direction during the third and fourth spectrums.

2. In event of a shut down of impulse testing due to breakdown or other reason a temperature-time spectrum shall be considered complete at the end of each 12,500 impulse cycles.

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3. If leakage occurs during impulse cycling that is great enough to hinder testing of other seals, then the system shall be shut down and an investigation made to determine which seal is leaking. Seal torque should be increased in increments of approximately 10% of original torque value or some reasonable squeeze value in case of face seals, until there is no evidence of leakage or the torque has reached 400% of the maximum vendor recommended value. If seal still leaks so as to hinder testing of other seals, it shall be blocked from the system.

II. Major Evaluation

A. Seal Selection

Only those types of seals satisfactorily completing the preliminary tests shall undergo final evaluation. These may be new seals or seals having undergone preliminary testing provided no evidence of physical change has appeared during preliminary testing.

B. Physical Check

The physical check shall be the same as outlined in Paragraphs I-A-1 through I-A-3.

C. Leakage and Proof Pressure

Leakage tests shall be the same as that outlined in Paragraphs I-B-1 through I-B-6.

D. Impulse Tests

1. Seals shall undergo a minimum of 200,000 pressure impulse cycles. Each pressure impulse cycle shall conform as nearly as possible to that shown in Figure II. Pressure impulse cycling shall be performed while the seals are being subjected to one of the temperature-time spectrums outlined in Figure I. The test shall be composed of 16 complete temperature-time spectrums. The pressure impulse cycling for the fourth, eighth, twelfth, and sixteenth temperature-time spectrums shall start when the seal and fluid temperature is at -65°F. All other pressure impulse cycling shall begin when the seal temperature is between 70°F

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and 100°F. Impulse pressure cycling shall be imposed in the reverse direction for those seals considered suitable for reversed pressure during the seventh through the tenth temperature-time spectrums (75,000 through 125,000 cycles).

2. Upon completion of each temperature-time spectrum (or each 12,500 cycles) starting with the eighth, each test seal (if applicable) shall be untorqued or otherwise have its sealing force relieved to simulate disassembly. The seal shall then be retorqued or loaded to the same level as when the spectrum was completed unless the seal was leaking upon completion of the spectrum.

3. In the event of shutdown of impulse testing due to breakdown or other reason, a temperature-time spectrum shall be considered complete at the end of each 12,500 impulse cycles. Reheat at times other than at beginning of a temperature-time spectrum will be done without cycling.

4. The procedure for leakage correction shall be the same as that in Paragraph I-C-3.

5. After completing 200,000 impulse cycles, each seal shall be examined visually and compared with the "as received" condition. All critical dimensions shall be checked again for determining any permanent set that may have occurred.

E. Burst Test

1. Subject each seal completing tests thus far to a hydraulic pressure of 10,000 psi from the inside diameter to out.

2. Repeat Paragraph II-E-1 for those seals considered suitable for reversed pressure, with pressure applied in opposite direction.

III. Supplementary Tests

A. Seal Selection

Each type and size seal completing the major evaluation shall be subjected to certain abuse tests. These seals, except those used in reverse pressure tests, shall be

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new seals not previously used in testing. The purpose of these tests is to determine the seals capability for sealing under abnormal squeeze and after improper previous use. Seals used in reverse pressure tests shall be those face seals not considered suitable for reversed pressure and having completed the major evaluation test.

B. Squeeze Tests

1. Install each seal in a suitable manifold or fitting and apply a torque or squeeze on the seal of approximately one-half the minimum recommended by the vendor. Repeat Paragraph I-B-2. If leakage occurs, repeat Paragraph I-B-3. Remove seal and check for permanent set, provided seal is capable of being removed.

2. Repeat Paragraph III-B-1 with torque or squeeze equal to the minimum recommended by the vendor provided this has not been reached already in Paragraph III-B-1.

3. Repeat Paragraph III-B-1 with torque or squeeze equal to maximum vendor recommended value.

4. Repeat Paragraph III-B-1 with torque or squeeze equal 1-1/2 times maximum vendor recommended value.

C. Re-Use Tests

1. Using the same fittings or manifolds used in Paragraph III-B-1, install a new seal in each and apply squeeze or torque equal to the minimum optimum value per previous tests. Repeat Paragraph I-B-2. If leakage occurs, increase toque in 10% increments, or some reasonable squeeze value in case of face seals, until leakage stops.

2. Release and retorque seals to the value obtained in Paragraph III-C-1 and repeat Paragraph I-B-2. Do this sequence until leakage occurs or sequence has been performed a total of ten times.

3. Repeat Paragraph III-C-2 ten times with torque applied each time doubled that applied in Paragraph III-C-1.

4. Repeat Paragraph III-C-2 until seal has been released and retorued a total of 25 times or leakage occurs.

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D. Reversed Pressure Tests

1. Each seal completing the major evaluation and which is not considered suitable for reversed pressure application shall have reversed pressure of 4000 psi applied for the purpose of determining effect on normal pressure sealing capability. Seals shall be torqued to optimum value per previous tests.

2. After the 4000 psi reversed pressure application, repeat tests per Paragraph I-B-2.

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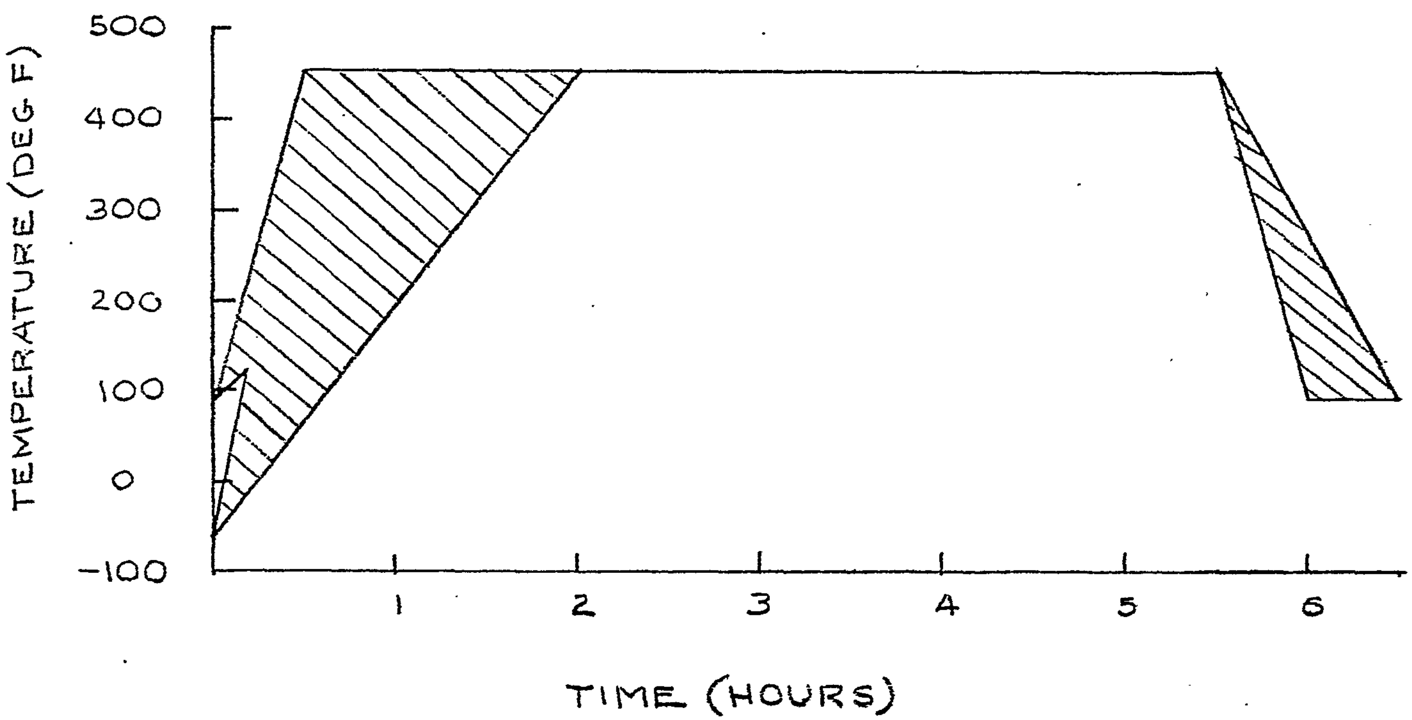
FIGURE I

MODULAR HYDRAULICS
TIME-TEMPERATURE SPECTRUM



NOTE:

CYCLING TO START AT MINUS 65 DEG F EVERY FOURTH TIME-TEMPERATURE SPECTRUM DURING MAJOR EVALUATION OF TEST SEALS

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FIGURE II
MODULAR HYDRAULICS
TYPICAL IMPULSE CYCLE

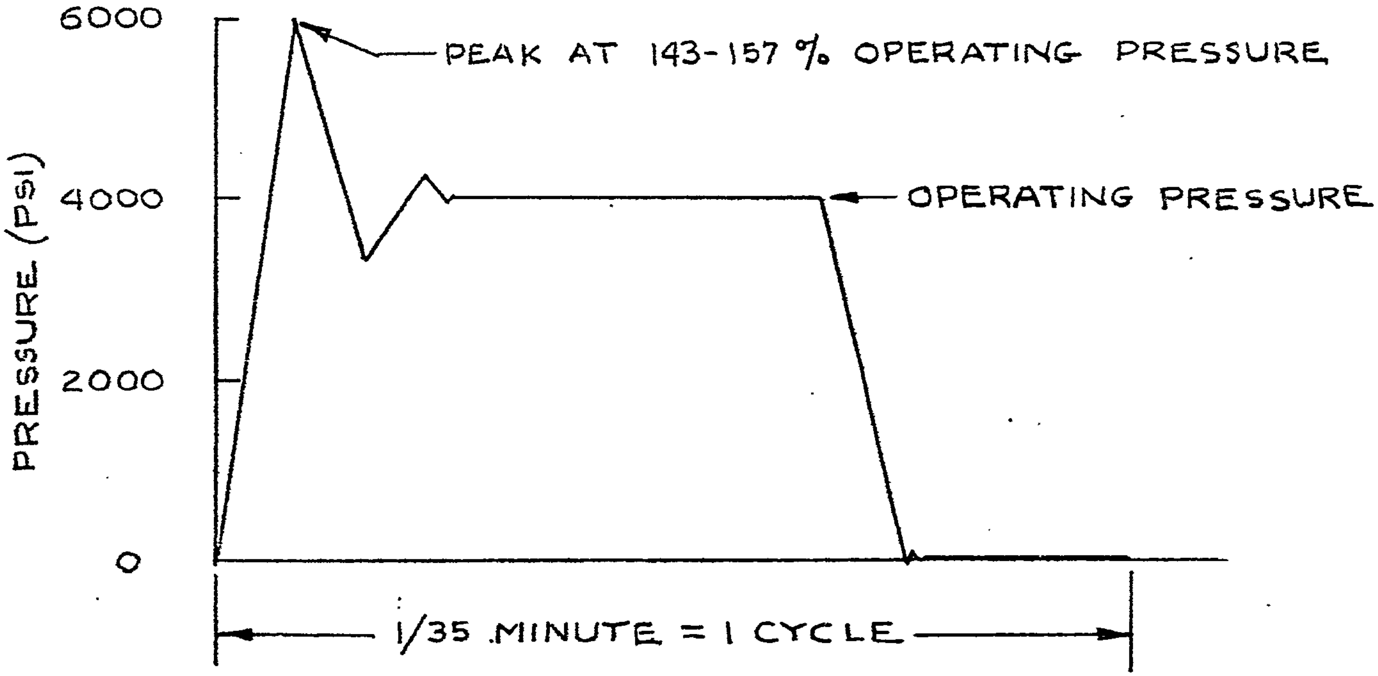

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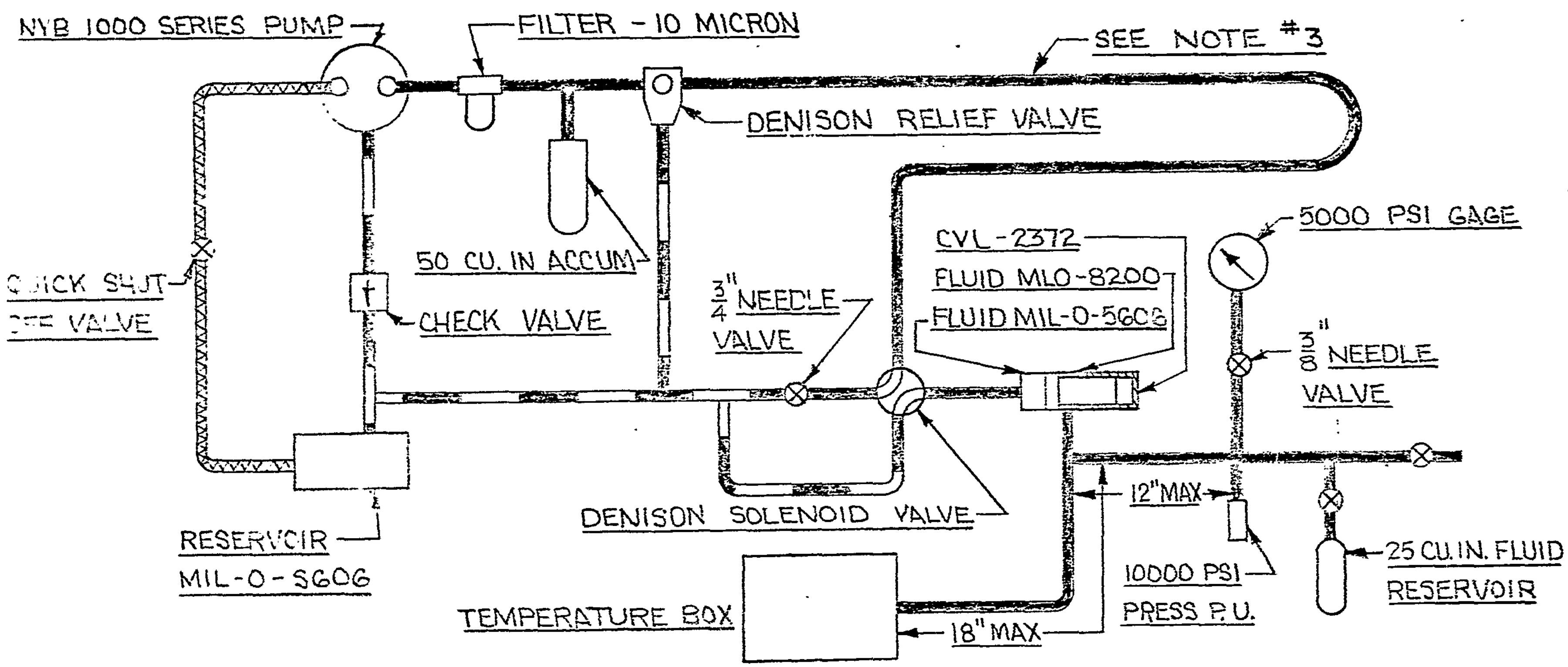
FIGURE III

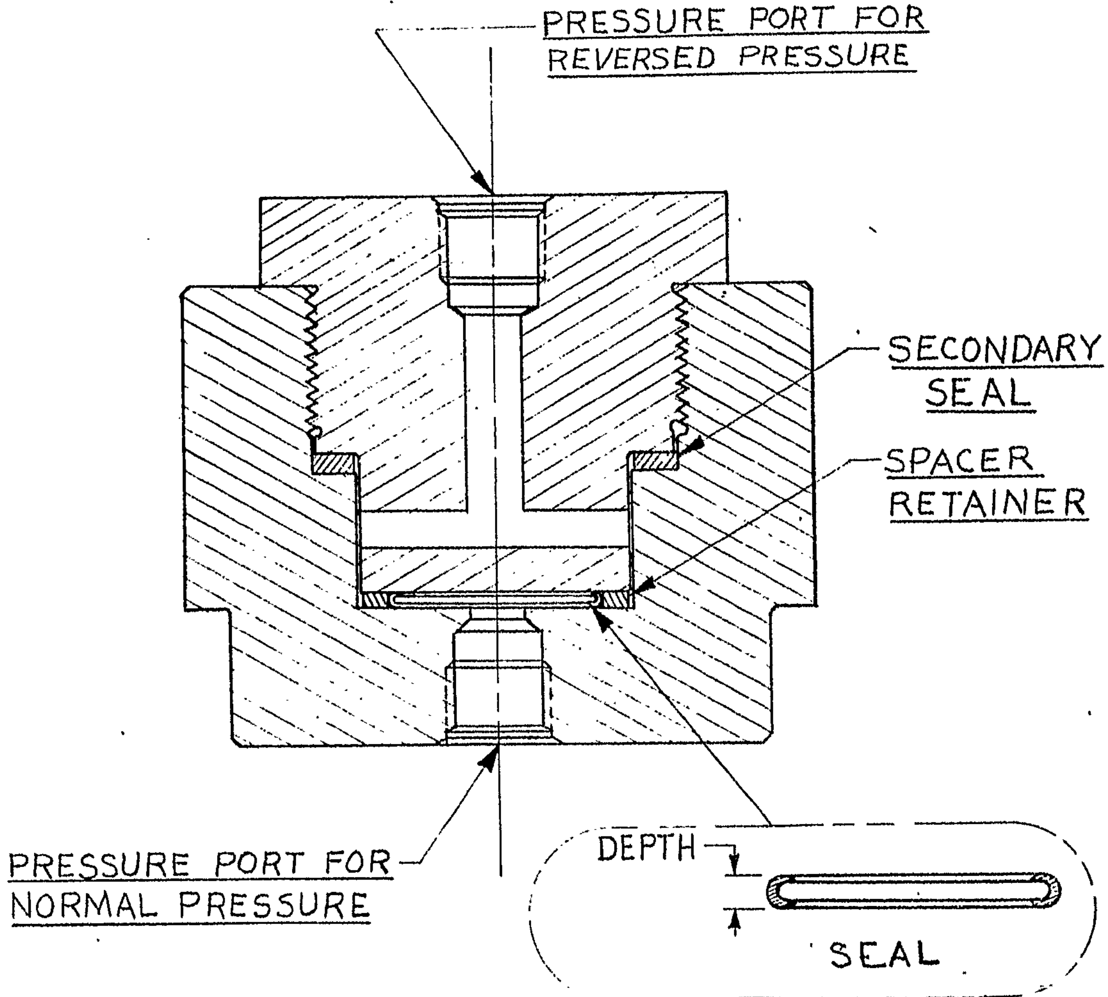
SCHEMATIC OF PRESSURE IMPULSE SET UP FOR TESTING STATIC SEALS



NOTES:

1. ALL CRES. STL. FITTINGS WITH WADC SEALS IN HIGH PRESSURE SIDE.
2. LINE SIZE: ALL $\frac{3}{8}$" EXCEPT SUCTION & RETURN TO BE $\frac{3}{4}$" HIGH PRESSURE LINES & PUMP PRESSURE LINE TO BE .035" WALL THICKNESS CRES. STL.
3. LINE FROM DENISON VALVE TO DENISON SOLENOID VALVE TO BE ONE CONTINUOUS UNBROKEN $\frac{3}{8}$" x .035 W.T. TUBE & A MINIMUM OF 20 FEET LONG. LINE MAY BE FOLDED BACK FOR SPACE CONSERVATION USING LARGE RADIUS TURNS.

TYPICAL METALLIC SEAL
TEST FIXTURE



APPENDIX II-4

PRELIMINARY SEAL TEST RESULTS:

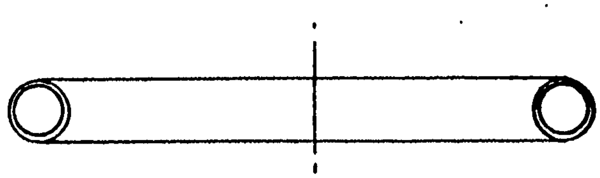
- ADVANCE SEAL
- AERO GASKET
- CADILLAC GAGE
- DOUBLE SEAL
- CHANGE VOUGHT X-SEAL
- HARRISON SEAL
- HI-CEAL
- NAVAN
- OMEGA
- RACO
- TORUSEAL
- WADC

ADVANCE SEAL TEST

The complete set of Advance Face Seals consisted of eight each of five sizes ranging from 0.375 to 1.750 inches I.D. Of the eight seals in each size, four seals were coated with a 0.001 - 0.002 inch thick silver plate, and four were coated with a 0.002 - 0.003 inch teflon coating.

Eight manifolds for testing these seals per CVC M.H.2 have been fabricated. Four manifolds contained silverplated seals and the other four contained teflon-coated seals. Each manifold tested two seals -- one given size and the next larger size, progressing from the smallest to the largest. Thus, each of the first four sizes of seals was tested as the small seal of each manifold.

Each manifold was dimensionally inspected to determine the actual amount of squeeze imposed on each seal. Each test seal was also dimensionally inspected as a reference for determining the amount of permanent set that occurs.



HOLLOW METAL O-RING

A total of sixteen Advance Seals (metallic O-rings) have undergone preliminary testing. Physical inspection, seal installation, static pressure and pressure impulse testing of the seals was performed. Leakage detection during pressure impulse cycling was provided by clear-sight tubes partially filled with hydraulic fluid and connected to the non-pressurized test seal cavity. Fluid leaking past the test seal displaced fluid upward in the sight glass which was in full view of the technician operating the test.

Testing of the seals listed in Table II-3 was performed in two groups. In the first group, one seal of each part number, the first listed, was initially set up in manifolds, static pressure tested and then pressure impulse cycled until failure or until a minimum of 50,000 p.i.c. (pressure impulse cycles) had been reached. The second group consisted of one seal of each part number, the second listed, and was tested in the same manner as the first group; however, any damaged cavity surfaces were first restored by very light sanding with No. 600 grit abrasive paper.

ADVANCE FACE SEAL TEST DATA

PART NUMBER	LAB S/N	SEAL O.D. Inches	SEAL Depth Inches	SEAL SQUEEZE Inches	CAVITY O.D. Inches	CAVITY Depth Inches	FINISH RMS	TOTAL P.I.C.	FAILURE
375-1-3-TC	1	.381	.034	.010	.381	.024	16-32	13,370	Yes
	2	.381	.035	.011	.381	.024	16-32	50,000	No
625-1-3-TC	2	.628	.034	.008	.631	.026	20-16	12,502	Yes
	3	.628	.035	.009	.631	.026	20-16	2,317	Yes
1000-1-3-TC	2	1.004	.036	.010	1.005	.026	16-48	52,000	No
	3	1.004	.038	.012	1.005	.026	16-48	33,890	Yes
1250-1-3-TC	2	1.254	.038	.008	1.256	.030	32-30	15,104	Yes
	3	1.255	.037	.007	1.256	.030	32-30	4,159	Yes
375-1-3-SP	1	.380	.034	.007	.386	.027	16-63	52,000	No
	2	.380	.035	.008	.386	.027	16-63	50,000	No
625-1-3-SP	2	.628	.035	.008	.630	.027	40-63	12,502	Yes
	3	.628	.035	.008	.630	.027	40-63	50,000	No
1000-1-3-SP	2	1.008	.038	.011	1.009	.027	40-125	15,104	Yes
	3	1.008	.037	.010	1.009	.027	40-125	50,000	No
1250-1-3-SP	2	1.255	.037	.010	1.255	.027	16-63	52,000	No
	3	1.261	.036	.009	1.255	.027	16-63	50,000	No

NOTES: (1) Seals manufactured from 0.035 inch dia x 0.006 inch wall thickness type 321 stainless steel, TC designating teflon-coated and SP designating silverplated.

(2) Finish check made by comparison with a surface checkboard conforming to ASA-B46, SAE, MIL-STD-10 and NAS 30 standards for designation and control of surface finish of precision machine parts.

TABLE II-3

After completion of the normal pressure impulse cycling, all seals of the second group which did not fail were set up for reversed pressure cycling. This included all the silverplated seals (indicated by SP in the part number) and one teflon-coated seal (375-1-3-TC S/N2). Before reverse pressure cycling, the plugs holding the seals in the manifolds were removed and the seals were examined and were found to be in good condition. The seals were not removed from the cavity. The plugs were then replaced and static pressure tests were performed with leakage occurring on two seals, 625-1-3-SP S/N3 and 1250-1-3-SP S/N3.

The 375-1-3 TC S/N2 seal completed 16,640 reversed p.i.c. without failure, being removed from the test only because of the failure of a secondary seal in the manifold. The two remaining seals, 375-1-3 SP S/N2 and 1000-1-3 SP S/N3, completed the target of 25,000 reversed p.i.c. without failure.

After completion of the impulse pressure cycling, all test manifolds were disassembled for examination of the seal and manifold mating faces. Damage to the manifold mating surfaces was generally insignificant. The teflon-coating on all teflon-coated seals had either been peeled away or had begun to do so. Some permanent set had occured in all seals and particular difficulty was encountered in removing the seals from their cavities. In most cases, the seals could not be removed without being damaged and they were not considered suitable for re-use.

The short life of the two 1250-1-3 TC seals was believed to be due to the low squeeze under which they were installed. The poor manifold surface finish may have caused early leakage of the 1000-1-3-SP S/N2 and the 625-1-3 SP S/N2 seals. Among the silverplated (indicated by SP in the part number) seals, it should be noted that only two failures occurred, which were perhaps justified in view of the poor surface finishes involved. Only two of the teflon-coated seals completed 50,000 p.i.c. without failure, therefore it may be concluded that the silverplated metallic O-ring may be suitable as a face seal under the conditions tested. Further testing to determine effects of various amounts of squeeze was not accomplished.

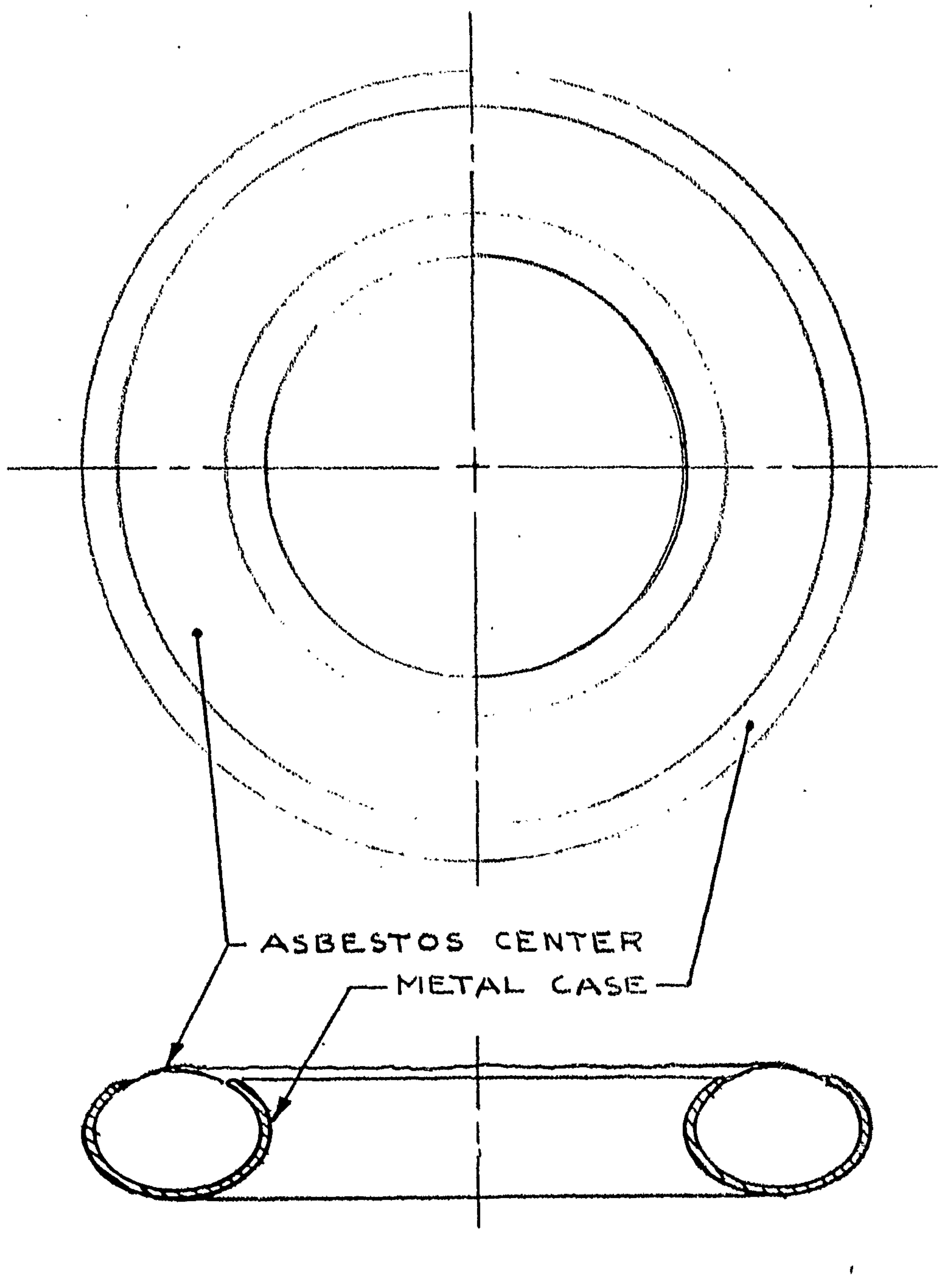
AERO GASKET SEAL TEST

The Aero Gasket Seal consists of a thin metal case enclosing an asbestos fiber center as shown in Figure II-15. A test manifold capable of applying various degrees of squeeze to the one test packing size that is presently available was fabricated.

Preliminary testing was performed on two of the four Aero Gasket sample seals using a test manifold similar to that shown in Figure II-16. The first seal (S/N1) was installed in the manifold without the spacer-retainer. The plug, having 5/16-24 UNF threads, was tightened to 75 in./lb. torque. The plug and seal were then removed and seal thickness and outside diameter were measured. A second seal (S/N2) was

FIGURE II-15

AERO GASKET SEAL



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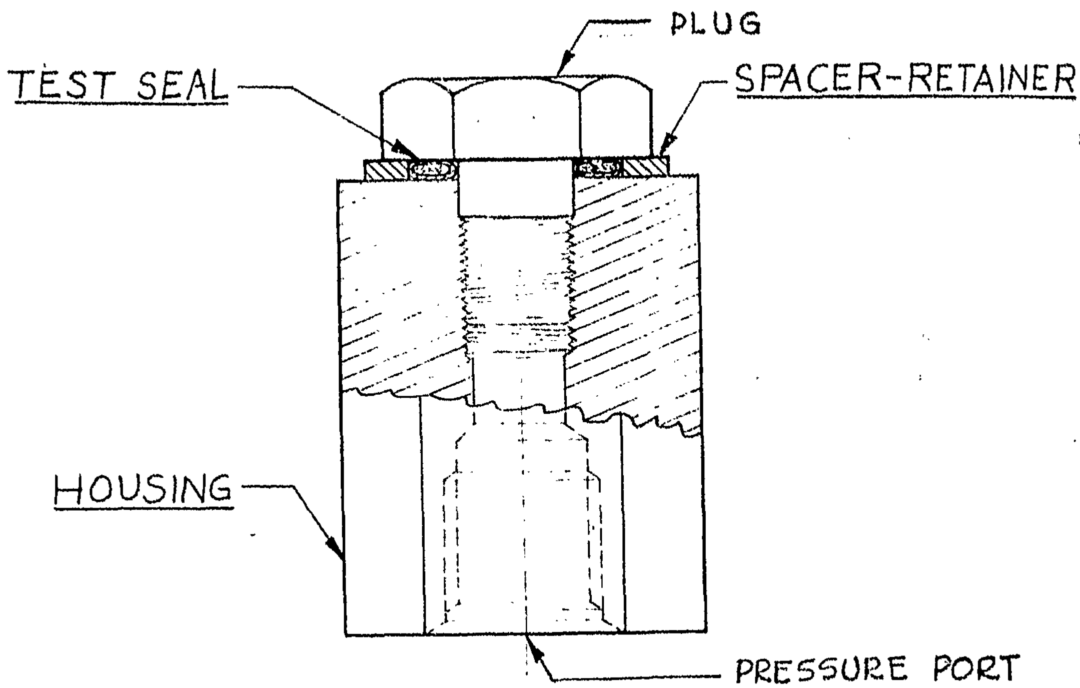


FIGURE II-16

then installed in the manifold along with a spacer-retainer. The spacer-retainer had a thickness and an inside diameter approximately equal to the thickness and outside diameter of the first seal after being tightened to 75 in./lb. torque. The purpose of the spacer-retainer was to control seal squeeze and to prevent failure of the seal under pressure loading. Static hydraulic pressures of 10, 4,000 and 6,000 psi were applied to the seal, however leakage occurred at 6,000 psi. Pertinent seal and spacer dimensions are given in Table II-4.

The seal and spacer-retainer were then removed and examined. Surface finish on mating surfaces of the manifold with the seal was approximately 16 RMS. The seal was again installed but with a new spacer-retainer which imposed 0.010 inch more squeeze on the seal. The seal then passed the static pressure tests to 6,000 psi satisfactorily. The seal was placed in the temperature box for pressure impulse cycling to 6,000 psi and 450°F in accordance with the pressure-time-temperature spectrum presented in CVC M.H.2. After 14,784 p.i.c. the seal began leaking and was removed from the temperature box and visually examined. The seal and surfaces appeared unchanged from the first installed condition and no reason for leakage was apparent. Upon removal, the seal depth was measured and found to be 0.0518 inch (measured depth new was 0.0890 inch). The seal was again installed in the same cavity. The seal was then subjected to 6,000 psi static hydraulic pressure for a period of 10 minutes without leakage occurring. When the squeeze was reduced by 0.001 inch, 6,000 psi static pressure caused leakage to occur.

TABLE II-4

AERO GASKET SEAL TEST

PERTINENT DIMENSIONS

ITEM	CONDITION	I.D. Inches	O.D. Inches	Depth Inches
SEAL S/N 1	As received	.321	.568	.090
	After tightening to 75 in./lb. torque	-	.585	.062
SEAL S/N 2	As received	.321	.568	.089
	See Note 1	.312	.570	.056
2nd SPACER-RETAINER	Used with S/N 2 Seal	.570	-	.051

NOTE: 1. Measurements taken after seal was removed from manifold but still retained in spacer-retainer. The 0.056 inch depth, an expansion of 0.006 after removal, is an indication of seal resiliency.

A new seal, S/N3, with a measured depth of 0.093 - 0.094 inch was installed in a 0.045 inch cavity which required a torque of 285 in./lb. On a 5/16-24 UNF plug, a static hydraulic pressure of 6,000 psi was applied to the seal without leakage occurring, however a 0.003 inch reduction of squeeze caused leakage to occur at 4,500 psi. The seal was tightened to the first squeeze value (0.045 cavity), static pressure tested to 6,000 psi without leakage, and installed in the controlled temperature box for pressure impulse cycling. The seal began leaking after 25,188 p.i.c. Pressure impulse cycling was performed in accordance with the pressure-time-temperature spectrum presented in the 3rd Quarterly Progress Report.

No further tests were accomplished on this seal design as tests to date indicate the resiliency of the seal to be so low as to render it unacceptable for use in a modular hydraulic system.

CADILLAC GAGE SEAL TEST

Preliminary tests were completed on four Cadillac Gage Company seals. Pertinent physical data for the seals tested are given in Table II-5 , and a view of the seal is shown in Figure II-17. All tests performed were in accordance with the general test procedure as presented in specification CVC M.H.2.

After installation of the seals in the test manifolds, each seal was subjected to static hydraulic pressures to 6,000 psi. There was no evidence of leakage and the seals were then placed in the temperature box for pressure impulse cycling to 6,000 psi and 450°F in accordance with the pressure-time-temperature spectrum. One seal (CGS41-0563) failed after 9,490 p.i.c. and another (CGS41-1438) failed after 47,356 p.i.c. The two remaining seals completed 50,000 p.i.c. without evidence of leakage.

Upon completion of tests, the seals were removed from the manifolds and the seal and manifold mating surfaces examined. Contact surfaces of the manifolds and seals completing 50,000 cycles were still smooth and showed no evidence of wear. The sealing edges of the CGS41-0563 seal were very jagged and one surface of the manifold had developed an erosion-like ring. The sealing edges of the CGS41-1438 were jagged and both mating surfaces of the manifold were just slightly eroded.

One seal completing the 50,000 cycles (CGS41-0813) was set up for reverse pressure testing. The seal was found to relieve at a pressure of 3,700 ± 100 psi and no further reverse pressure tests were performed. The squeeze and cavity dimensions were the same as for normal pressure tests.

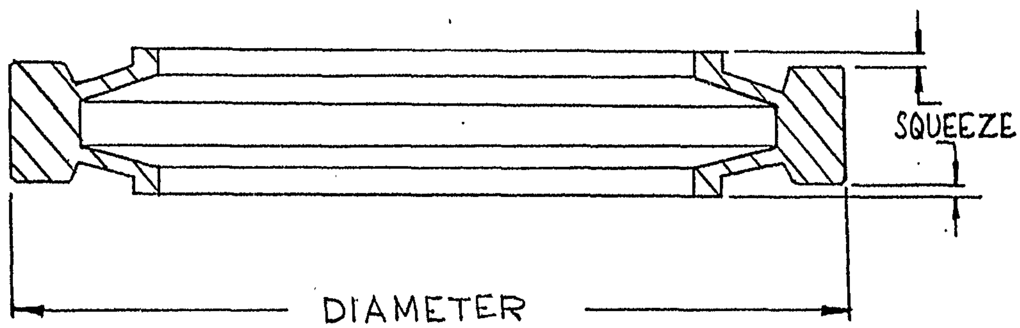
Squeeze on the Cadillac Gage seals is controlled by the seal shoulder as shown in Figure II-17 . Vendor tolerances on the seal range from 0.002 inch to 0.020 inch; however, of 20 seals measured, the squeeze was between 0.004 and 0.006 inch. Because it has been determined that seal cavity machining tolerances will cause squeeze to vary 0.008 inch between cavities, this seal was eliminated from further consideration.

TABLE II-5

CADILLAC GAGE SEAL TEST DATA

PART NUMBER	SEAL			CAVITY	
	I.D. Inches	O.D. Inches	Squeeze Inches	O.D. Inches	Finish RMS*
CGS41-0563	.299	.564	.005	.576	32
CGS41-0813	.547	.815	.005	.826	32-50
CGS41-1188	.919	1.188	.006	1.201	32
CGS41-1438	1.171	1.438	.005	1.451	32

* Finish check made by comparison with a surface checkboard conforming to ASA-B46, SAE, MIL-STD-10 and NAS 30 standards for designation and control of surface finish of precision machine parts.



CADILLAC GAGE SEAL
FIGURE II - 17

DOUBLE SEAL TEST

Static leakage tests were performed on two sample sets of Double Seals supplied by the Double Seal Ring Company, Fort Worth, Texas. The seal consists of two parts, an inner steel expander and an outer steel ring, both having a rectangular cross-section. A view of the seals and the test manifold are shown in Figure II-18.

The outer diameter of the dummy piston on which the seals were installed measured 1.497 inches and the cylinder bore diameter was 1.501 inches. Surface finish of the cylinder bore measured 1.5 to 2.5 RMS. The surface finishes of the seals and the lands of the dummy piston could not be measured but looked comparable to that of the cylinder bore.

Using MLO-8200 hydraulic fluid at room temperature, static hydraulic pressure of 100 psi and 4,000 psi was applied to the center port of the test manifold with the dummy piston and seals installed as shown. Fluid leakage past each seal while under pressure was channeled through the end ports and into a graduate. Leakage rate at 100 psi was 10 cc/min. and 28 cc/min. for each seal, respectively; and at 4,000 psi, it was 22 cc/min. and 28 cc/min. No further testing of this seal design is planned since its leakage rate is so high as to render it impractical for use in the modular program.

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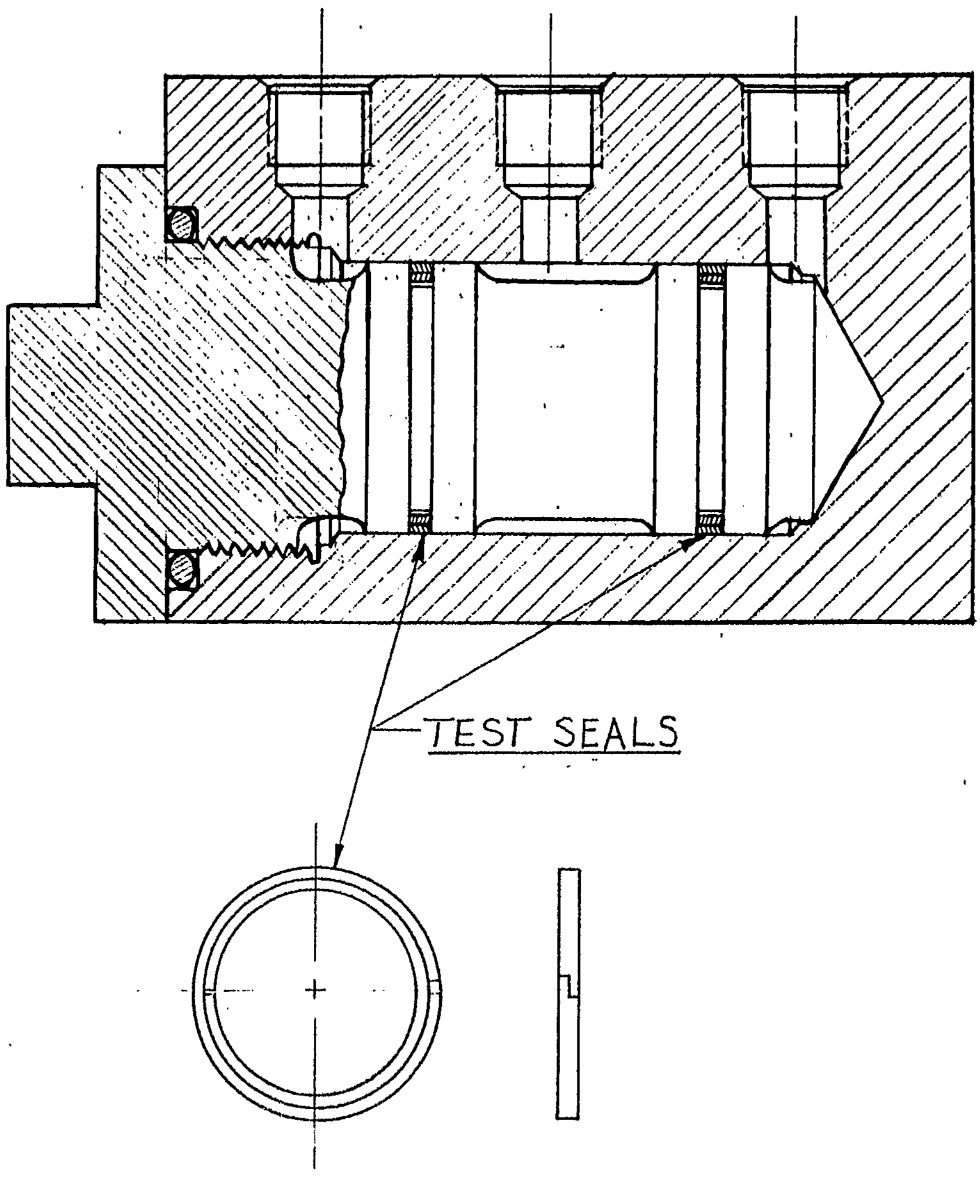
FIGURE II-18

DOUBLE SEAL IN TEST MANIFOLD



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CHANCE VOUGHT X-SEAL

Installation and static hydraulic pressure tests were performed on one Chance Vought CVP-4242-3 X-Seal which is shown in Appendix 1. The seal is composed of two formed pieces brazed together to form a ring having an X-shaped cross-section. The seal design is intended to be capable of withstanding both normal and reversed pressure applications.

The tests included installation of the seal in a test housing, squeezing the seal various amounts and performing static hydraulic pressure tests at each value of seal squeeze. During each test, static hydraulic pressures were applied first to the seal I.D. and then to the seal O.D., slowly raising the pressure to 6,000 psi each time or until leakage occurred. The seals free depth was carefully checked before and after each installation. Physical installation of the seal for tests is shown in Figure II-19 and results of tests are presented in Table II-6. The photograph, page 143 , shows the seal after it had collapsed radially due to the 6,000 psi reversed pressure. As shown in the figure, the seal was not closely supported in a radial direction during testing.

Tests results indicated poor sealing performance of the seal; however, in this particular seal, this was a result of a tolerance problem in the manufacture of the seal and not necessarily a result of the seal design. The seal did not leak with 6,000 psi hydraulic pressure applied in the reverse direction even though the squeeze on the outer lips of the seal was only 0.007 inch. This was also true after the seal had collapsed during the application of reverse pressure. The seal tested had a measured depth which varied from 0.0505 inch to 0.0700 inch, whereas the design specified a depth of 0.064 ± 0.005 inch.

Based upon the partially successful test results obtained on this seal and its possibilities of low cost, a decision was made to fabricate four more of these seals for additional testing. An attempt was made to secure better quality seals by holding the brazing fixtures to closer tolerances.

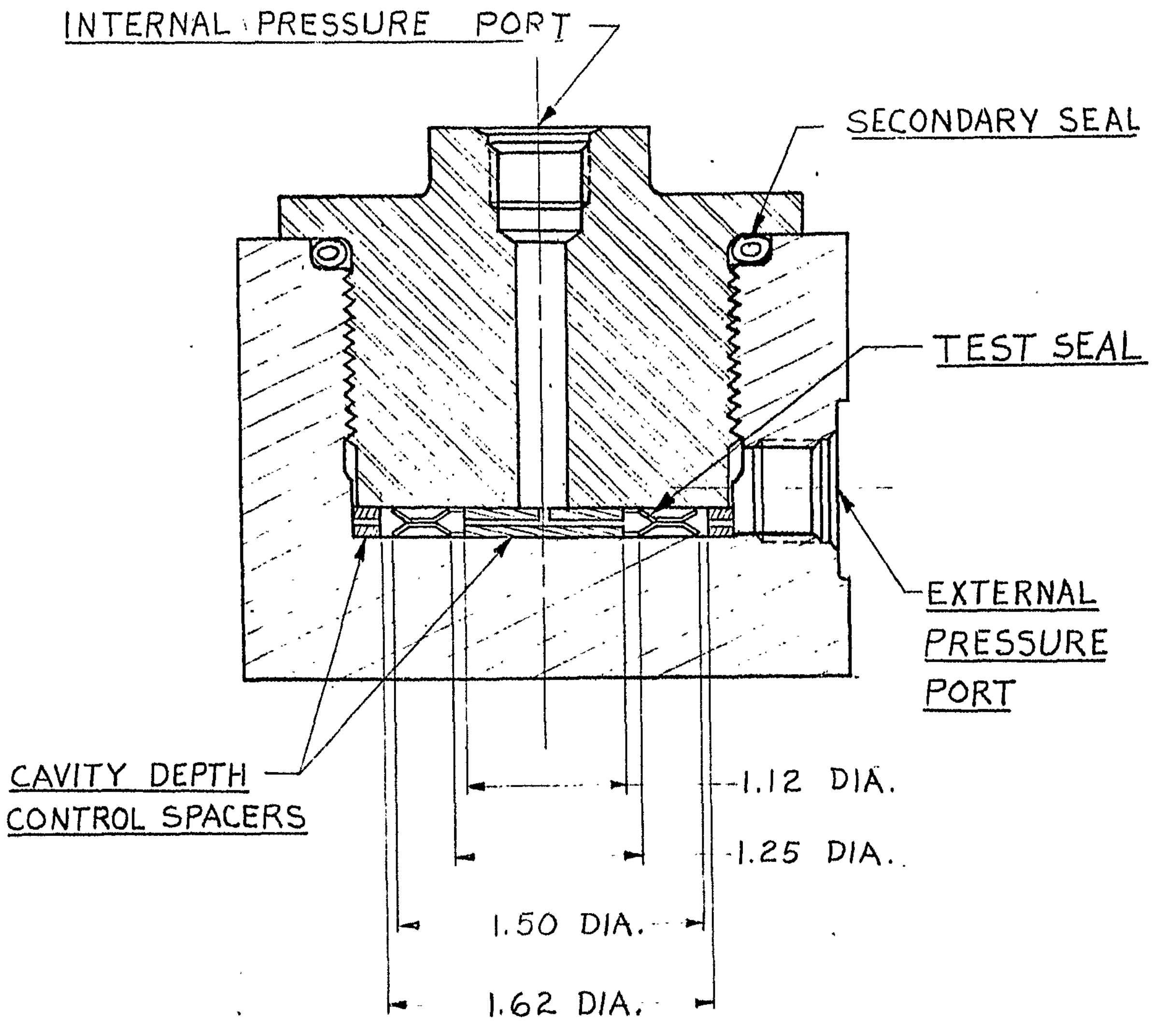
Hydraulic tests were performed on four additional CVP-4242-3 "X" seals. All tests were performed using a test housing as shown in Figure II-19 . Test details and results for the four seals tested are shown in Table II-7.

Sealing capability for the "X" seal design was found to be very good at room temperature; however, tests performed to determine its effectiveness at elevated temperatures were not encouraging. The seal was found to require close diametral support to prevent yielding of material under hydraulic proof pressure of 6,000 psi. The force versus deflection curve shown in Figure II-20 points out that deflection forces are completely unacceptable. It is apparent that considerable development will be necessary to perfect the "X" seal. Since the modular program did not include funds for such a program, the "X" seal was dropped. A design drawing of the "X" seal is seen in CVC 4242.

FORM NO. 38

FIGURE II-19

CHANCE VOUGHT X-SEAL IN TEST HOUSING



REVISION. DATE:

J.J.W
12-15-59

TABLE II-6

CHANCE VOUGHT X-SEAL TEST RESULTS

CAVITY DEPTH (Inches)	SEAL DEPTH* (Inches)				PERFORMANCE DURING PRESSURE APPLICATION***	
	Inner		Outer		Pressure to I.D.	Pressure to O.D.
	Min.	Max.	Min.	Max.		
**	.0505	.0555	.0630	.0700	-	-
.064	.0502	.0555	.0628	.0668	Would not seal	Would not seal
.056	.0515	.0559	.0555	.0562	Sealed to 900 psi	Sealed to 6,000 psi, but collapsed radially
.048	.0479	.0494	.0487	.0500	Would not seal	Sealed to 6,000 psi, although previously collapsed.

NOTES: * Depth measurements were made in 12 marked locations each around the circumference of the seal's inner and outer lips before testing and after removal from each test set-up with cavity depths shown. Maximum and minimum locations were not always repetitive.

** Depth measurements are for new seal as received.

*** All tests were performed with fluid and ambient temperatures of 75 ± 10 degrees Fahrenheit.

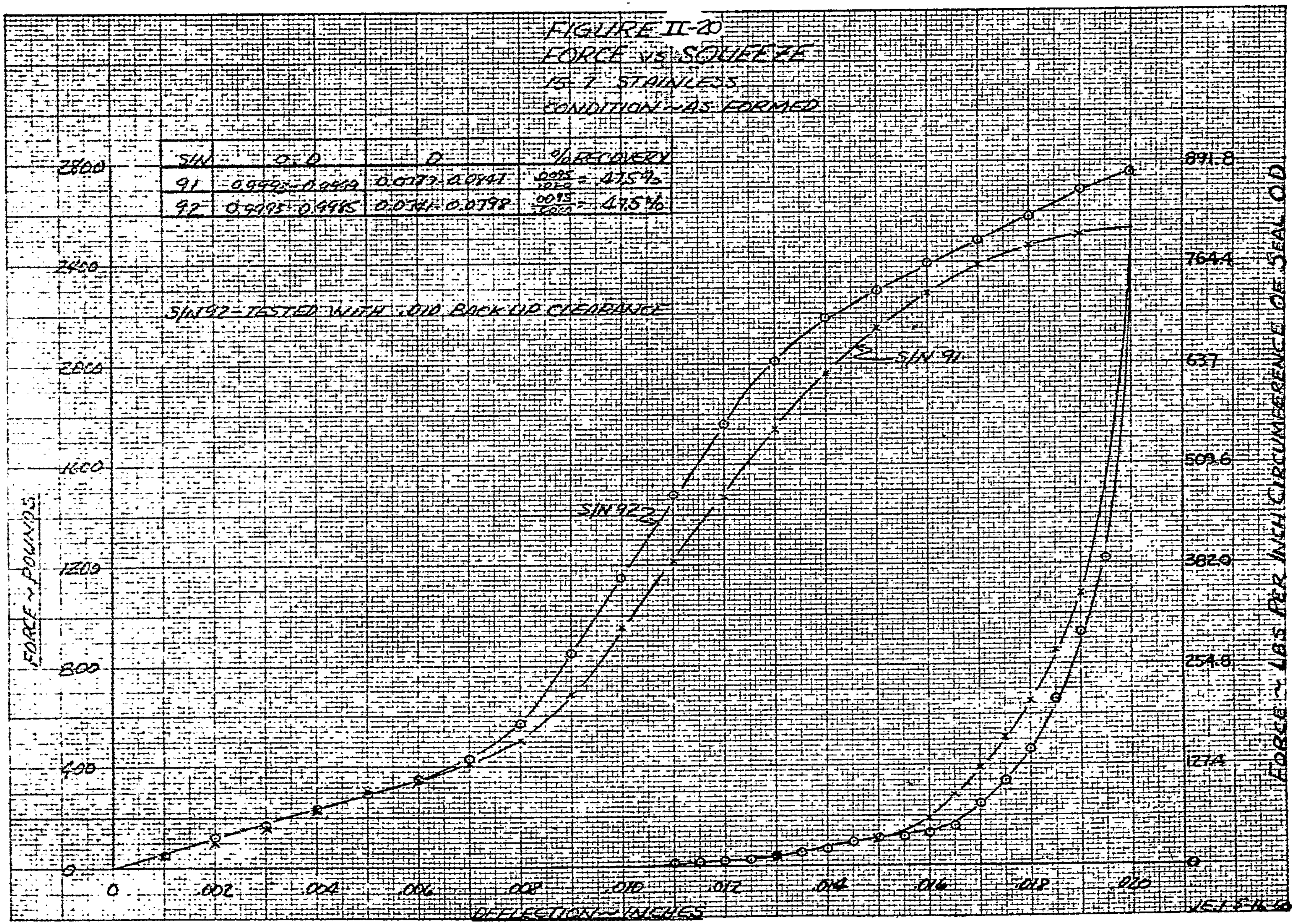
PHOTOGRAPH V-1 - A VIEW OF [illegible]V-4242-3 [illegible] SHOWING FAILURE RESULTING FROM APPLICATION OF 6000 [illegible] REVERSE PRESSURE

[redacted]

TABLE II-7

CVP-4242-3 "X" SEAL TESTS

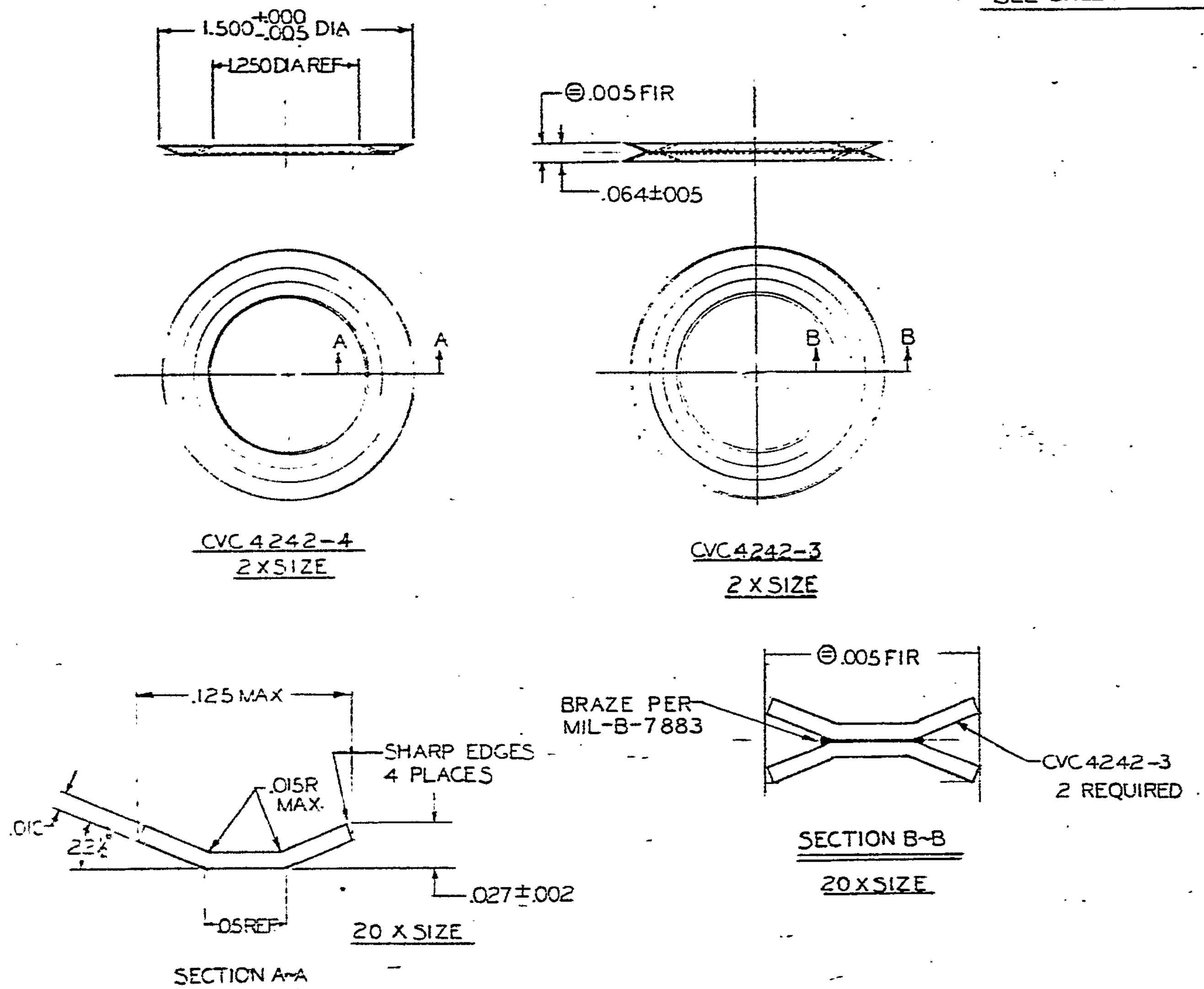
S/N	SEAL DEPTH (Inches) Outer Lip	Inner Lip	SEAL OD-In.	ID-In.	CAVITY (Inches) O.D.	I.D.	Depth	TEST DETAILS
2	.0618-.0649	.0595-.0640	1.50	1.25	1.62	1.13	.056	On application of internal pressure, seal leaked at pressures in excess of 700 psi at room temp. External pressure to 6,000 psi was applied at room temp. without leakage. On removal, found seal collapsed against cavity ID. Installed seal in .048 x 1.13 x 1.62 in. cavity. Applied 6,000 psi internal pressure without leakage. Removed seal and found it had partially returned to its original circular shape due to internal pressure. Installed and removed seal 4 times in .048 inch cavity. Applied 6,000 psi internal pressure each time w/o leakage. Made final installation and applied 6,000 psi reversed pressure. High leakage occurred up to 100 psi with 0 leakage at 6,000. Seal depth after test measured .048-.050 in.
3			1.50	1.25	1.62	1.25	.052	Applied 6,000 psi internal pressure; removed seal; found it had yielded and expanded to cavity OD. Installed again in same cavity; applied 6,000 psi internal pressure at room temp. w/o leakage. After stabilizing at 450°F seal would not seal against internal pressure. After cooling to room temp., internal sealing was effected again to 6,000 psi.
4	.0608-.0665	.0617-.0638	1.50	1.25	1.62	1.25	.060	Slowly applied 6000 psi reverse pressure at room temp. and again after stabilizing at 450°F. A total of 3120 reverse pressure impulse cycles were imposed on seal at temp. to 450°F. No leakage or failure to the seal occurred during any of the tests.
6	.0658-.0689	.0582-.0597	1.50	1.25	1.51	1.26	.048	Slowly applied internal pressure of 6000 psi at room temp. w/o leakage. After stabilizing at 450°F excessive leakage occurred at low internal pressure. Removed seal and measured outer and inner lip depths to be .0468-.0483 in. and .0497-.0505 in., respectively.



10 X 10 TO THE CM. 359-14
KEUFFEL & ESSER CO. MADE IN U.S.A.

SEE SHEET ONE FOR NOTES

REVISIONS
A ADDED SHEET 2, TO PROVIDE -3 AND -4.



REQ'D	ZONE	PART NUMBER	NOMENCLATURE	DIA	THICK	WIDTH	LENGTH	MATERIAL	MATERIAL SPEC	LBS/100 IN.	FINISH	WT
2		CVC4242-4	WASHER	2.0	.01	—	—	321 SHEET	MIL-S-6721 OR EQUIVALENT			
		CVC4242-3	SEAL									

TOLERANCE UNLESS OTHERWISE NOTED: DETAIL ±.005, ASSEMBLY ±.010, ANGULAR ± ½°

SCALE: NOTED

X SEAL
HIGH PRESSURE HIGH TEMPERATURE
MODULAR HYDRAULIC

CVC 4242 | SHEET 2 | REVISION A

CVC 4242 A

HARRISON SEAL TEST

A total of 29 Harrison metallic boss seals (Figure II-21) underwent testing. All seals other than sample seals were visually and dimensionally checked for conformance with the manufacturer's specifications. All seals were installed on MS-type or laboratory-made hydraulic fittings which conformed to MS standards. They were then subjected to a static leakage test at 10, 4,000 and 6,000 psi before undergoing p.i.c. (pressure impulse cycle) testing to 6,000 psi at approximately 35 cycles per minute. Sample seal testing did not conform to any particular temperature-time-spectrum, however almost all p.i.c. testing was performed at 450 ± 10°F. The temperature-time spectrum during p.i.c. testing of all other seals was in accordance with the General Test Procedure for Metallic Seals, Appendix II-3 except for certain time lapses caused by malfunctions. Detection of small amounts of leakage during p.i.c. at elevated temperature was very difficult. The methods employed in detecting leakage were: (1) checking for smoke in heating air escaping from temperature box; (2) change of captive fluid volume contained in the test seal fittings and the intensifier as indicated by an indicator rod; and (3) visual observation of seals and fittings after completion of a test run at high temperature. A view of the set-up for p.i.c. testing is shown in the photograph, page 154. This photograph shows the initial set of sample seals with companion fittings in the temperature box. A recording of a pressure impulse-time cycle made during p.i.c. testing is shown in Figure II-22, which is typical for all p.i.c. testing. The extent of testing, initial torque, and final torque values for individual test seals is shown in Table II-8.

Sample Seals

Sample seals were received prior to the formal test seals and are identified by the letter "S" in their serial number. Tests were begun on these samples before the formulation of a formal test plan in order to gain preliminary information which would be useful in writing a test plan. Since the sample seal testing did not initially conform to the formal test plan and since they occupied very little space in the temperature box, they were allowed to continue being subjected to p.i.c. testing without regard to a specific total number of cycles as an objective.

All sample seals completed a total of 176,600 p.i.c. Initially they were torqued to the minimum torque values as specified by specification CVC 12-182a. After 6,804 p.i.c., the -12 seal began leaking. The seal was removed and visually inspected. It required approximately 50 in.-lb. torque to remove the seal, whereas it had been installed at under 500 in.-lb. torque. It was noted that the teflon had begun to peel from the seal. The seal was installed again and torqued again to 500 in.-lb. All other sample seals were torqued to the original torque values with some rotation occurring in all cases. Pressure impulse cycling was then resumed with no leakage until after a total of 17,966 p.i.c. when one -6, the -10 and -12 seals were found to be leaking. They were torqued again to the original torque values and leakage stopped.

After completing 20,100 p.i.c., all seals were released and retorqued to the minimum torque values recommended by the Harrison Seal Company; i.e., 40, 60, 132 and 190 in.-lb. for the -4, -6, -10, and -12 seals, respectively. This information was not available prior to this time. Testing was resumed and at the end of 26,257 p.i.c. The -12 seal was leaking slightly. Retorquing to 190 in.-lb. stopped leakage. After 28,478 p.i.c., both the -10 and -12 seals were leaking. Leakage was stopped by retorquing to 132 in.-lb. and 190 in.-lb., respectively, with some rotation occurring. At the end of 29,359 p.i.c., the -12 seal was again leaking. The seal was again torqued to 190 in.-lb. with resultant rotation and stoppage of leakage. At the end of 30,108 cycles, the -10 and -12 seals were leaking again. Leakage was stopped again by torquing to 132 and 190 in.-lb., respectively. Also at this point all other seals were retorqued to the minimum Harrison-recommended torque values with rotation occurring in all cases. All companion fittings to the seals were then safety-wired to prevent their backing off. Continuing testing, 131,677 p.i.c. were completed without leakage on the -4 and -6 sizes; however, some seepage continued to occur on the -10 and -12 seals even though the torque was ultimately raised to 600 and 800 in.-lb., respectively. After a total of 163,000 p.i.c., one -6 and the -12 seal were again found to be leaking slightly. Leakage was stopped by tightening each to a value in excess of the AND 10064 value. No further leakage occurred and testing was stopped after 176,600 p.i.c.

<u>Test Seals - Set 1</u>

The first full set of formal test seals included one each of a teflon-coated and a goldplated seal in a -6, -10, -16, -20, and -28 size. The seals were installed on fittings torqued to the minimum value recommended by the Harrison Seal Company and then safety-wired. All seals passed the static leakage test under the initial torques with exception of the -16 goldplated seal. Torque on this seal was ultimately raised to 1,020 in.-lb. without effecting a seal. The MS 21915-16-10C fitting in which it was installed was found to have a slight dent in the chamfer area of the boss which mated with the seal. The fitting was replaced. The seal then required a torque of 840 in.-lb. to effect a seal at 6,000 psi. A lower torque value perhaps would have been sufficient had not the seal already been subjected to a higher torque on the previous fitting.

After 4,600 p.i.c., all seals had begun leaking with the exception of the -6 and -10 goldplated seals. All leaking seals were retorqued in an effort to effect a seal. Seal size, initial torque, torque causing rotation to begin, and final torques in in.-lb. listed in that order are as follows:

- 6CR	60,	25,	70	-28CR	410	----,	1008
-10CR	132,	90,	150	-28AG	1170,	1290,	1920
-16CR	237,	---,	280	-20AG	835,	648,	1044
-20CR	293,	160,	330	-16AG	675,	648,	1332

All seals except the -28CR and -16AG stopped leaking under the above torques. All seals were then removed and both the seals and fittings were visually examined. The teflon and goldplating were found to be peeling from most seals as shown in the photographs, pages 157 &. 158.

Also, the inner sealing edges of the seals and the surfaces of the fittings mating with the inner sealing edges of the -28CR and -16AG seals were found to have become slightly eroded similar to that found in other fittings.

Two additional -20 seals (a teflon-coated and a goldplated seal) were tested through a total of 50,000 p.i.c. Both seals were initially installed on fittings, safety-wired and torqued to 297 in.-lb. each. Both seals passed the static leakage test satisfactorily. After 20,860 p.i.c., both seals had begun leaking. It was discovered at this point that the minimum torque on the goldplated seal initially should have been 835 in.-lb. per the Harrison Company. The seal had been torqued inadvertently to the same value as required for teflon-coated seals (Table II-9). The goldplated seal was then torqued to 835 in.-lb. and the teflon-coated seal was retorqued to 297 in.-lb. Both seals leaked, however, when subjected to the static pressure leakage test. Upon disassembly, the faces of both MS 21916-20-16C fittings on which the seals were installed were found to contain eroded annular grooves which mated with the inner sealing edge of the seal. The faces of both fittings were reworked by sanding and restored to their original finish. Since the teflon on the teflon-coated seal was now in poor condition (similar to that in photograph, page157), it was completely stripped away with a soft brass tool. Both seals were then installed on the refinished fittings and each torqued to 835 in.-lb. The seals then passed the static leakage test satisfactorily.

Pressure impulse testing was resumed. At the end of 36,847 p.i.c., both seals were again leaking. Seals were alternately torqued and static pressure tested until a torque of 1,380 in.-lb. was reached. At this point, the goldplated seal passed the static leakage test satisfactorily, however the teflon-coated seal continued to seep slightly when pressurized to 6,000 psi. Torque was increased no further and pressure impulse testing was continued until a total of 50,000 p.i.c. had been completed. Some seepage was noted to have occurred on both seals at various times during the completion of the pressure impulse cycling. The seals were removed from the fittings and both the seals and fittings were visually inspected. There were eroded annular grooves in two of the fittings. The seals were similarly eroded but to a lesser degree.

The results of tests on this particular set of seals indicated that once leakage under high pressure has occurred, effecting a seal becomes difficult if not impossible due to erosion of the sealing surfaces caused by leakage. It was decided at this time to discontinue tests on this set of seals and set up a new set of seals on new or refinished fittings and torque them to a much higher initial value. It was felt that a higher torque perhaps would enable the seal to continue sealing even after extrusion of the teflon finish or yielding of the gold finish.

Test Seals - Set 2

The second full set of formal test seals included one each of a teflon-coated and a goldplated seal in a -6, -10, -16, -20 and -28 size. The seals were installed on fittings, torqued to approximately the highest values specified by AND 10064 (see Table II-9), and then safety-wired. The seals completed the static leakage check satisfactorily. All seals other than the -16 and -28 teflon-coated seals were subjected to 61,450 p.i.c. without leakage. After 8,660 p.i.c., the -16 and -28 teflon-coated seals were found to be leaking.

The -16 teflon-coated seal was retorqued to 1,368 in.-lb. after which no further leakage occurred during the remainder of the 28,591 p.i.c. In retorquing the seal, it was noted that rotation of the fitting began at approximately 648 in.-lb. and turned approximately 20° before reaching the 1,368 in.-lb. level. Although this seal was not visually examined after leaking, it is believed that the loss of squeeze on the seal was caused by loss of the teflon coating on the seal's sealing edge.

The -28 teflon-coated seal was torqued to 3,360 in.-lb. without effecting a seal under static pressure of 6,000 psi. The seal was then removed and visually examined. No cause was apparent for its failure to seal. A new -28 teflon-coated seal was then installed in place of the leaking seal. This seal was torqued to the AND 10064 torque value also. It passed the static leakage test satisfactorily. After 7,090 p.i.c., this seal also began leaking. The seal was removed and again no apparent cause could be found for the leakage. As with the previous seal, leakage had occurred between the seal and the same -28 laboratory-made union. To eliminate the possibility of the union being at fault, the union-to-seal mating surface of the union was reworked to a finish considerably better than the minimum finish required by the Harrison Company.

Another new -28 teflon-coated seal was then installed and torqued to the AND 10064 torque value. This seal passed the static leakage test satisfactorily; however, after setting overnight and prior to pressure impulse cycling, a static pressure test was again run on the seal. The seal began leaking as the pressure was raised to approximately 3,000 psi. Increasing the pressure to 6,000 psi would sometimes effect a seal and sometimes would not. The seal was removed, seal and mating surfaces were visually examined, and no cause for leakage could be found. By marking the seal and fittings and locating the seal in various rotated positions between the fittings, it was noted that the point of efflux of fluid during pressure application was always in the same spot relative to the seal. Closer visual examination of the seal in the leakage area still revealed no apparent cause for leakage. No further testing was performed on this size of teflon-coated seal.

Special Test Seal

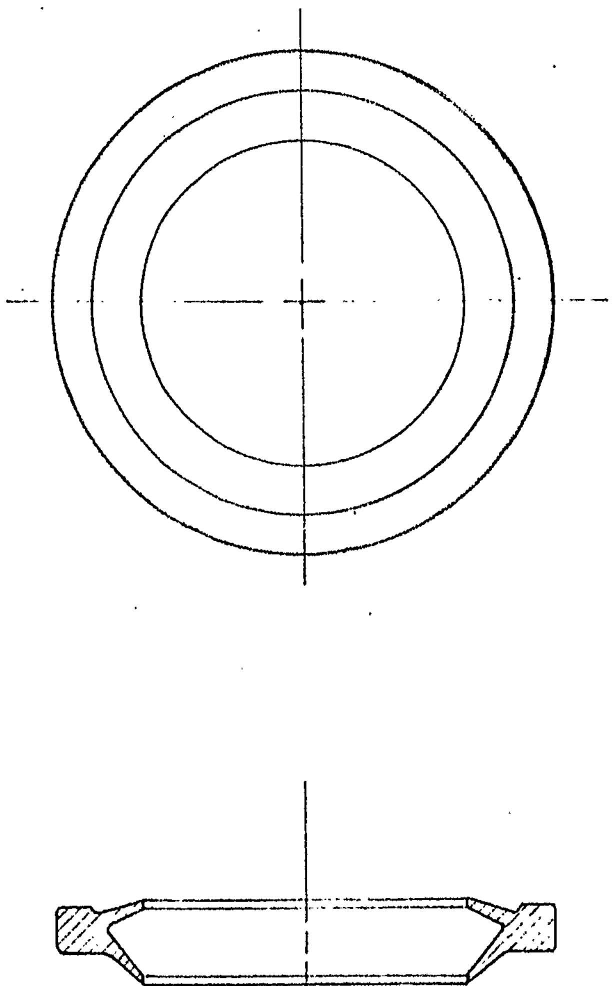
One -16 teflon-coated seal was received from the Harrison Seal Company which was already installed in their fittings and was tightened to 260 in.-lb. torque. The seal was tested at static pressures of 10,400 and 6,000 psi with no evidence of leakage. The seal and fittings were installed in the temperature box and pressure impulse cycling was started. After 12,500 p.i.c. at temperatures to 450°F, the seal was found to be leaking. The leakage rate was small and the cycling was continued. After a total of 22,213 p.i.c., the seal and fittings were removed because the leakage rate had become excessive. The seal was subjected to slowly increasing pressure at room temperature. At approximately 4,000 psi, the seal started to leak. No physical changes were made to the seal or fittings and no further tests were performed.

Conclusion

On the basis of the above tests, the results indicate the following: (1) Harrison torque values for teflon-coated seals are too low, (2) Goldplated seals are superior to teflon-coated, (3) Seals are more effective at room temperature than at elevated temperatures, (4) Seals that leak at one pressure sometimes seal at a high pressure, and (5) Stopping leakage becomes more difficult the longer a seal is pressure cycled.

FIGURE II-21

HARRISON MANUFACTURING COMPANY
12100 SEAL FOR AND 10050 BOSS



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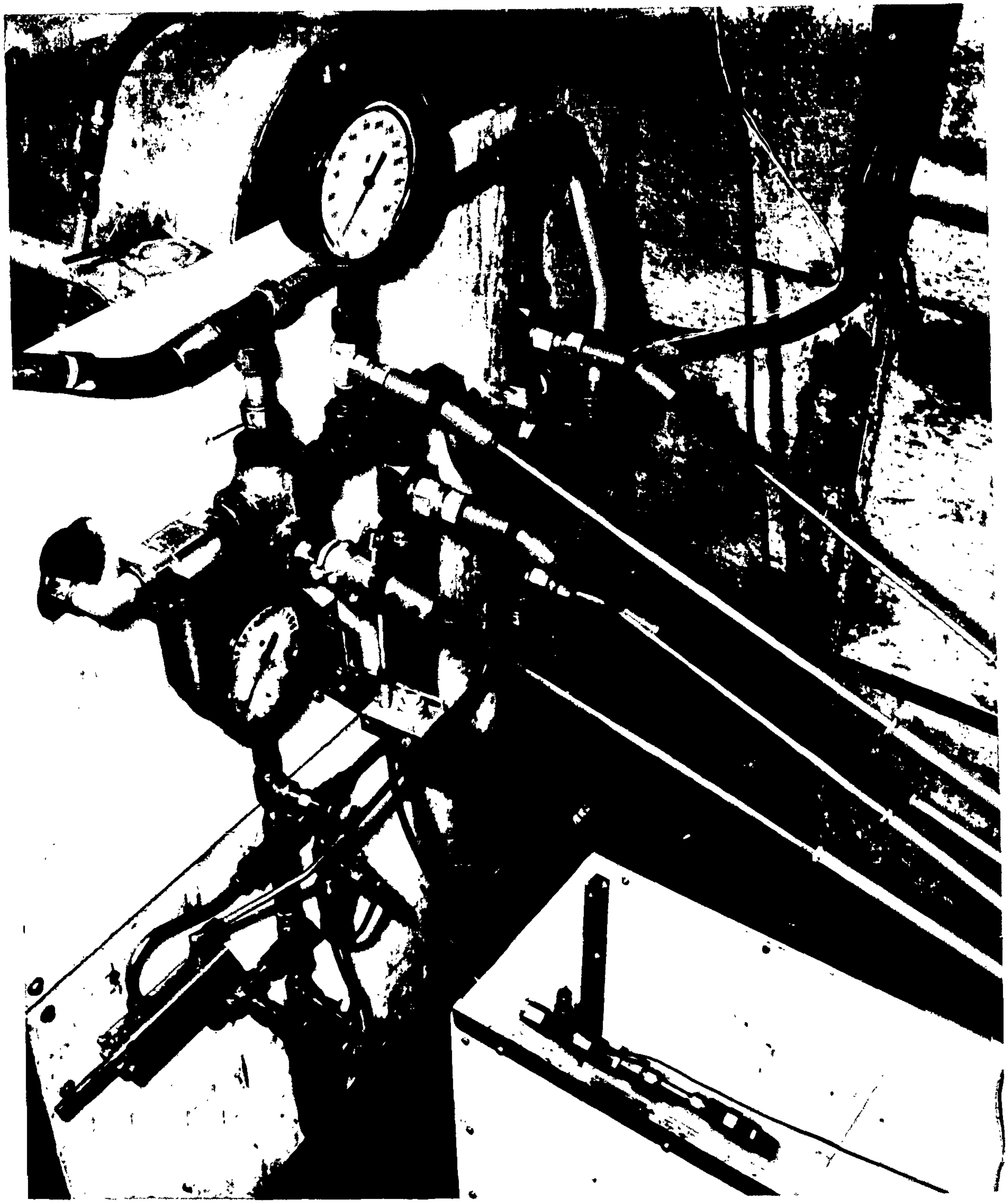
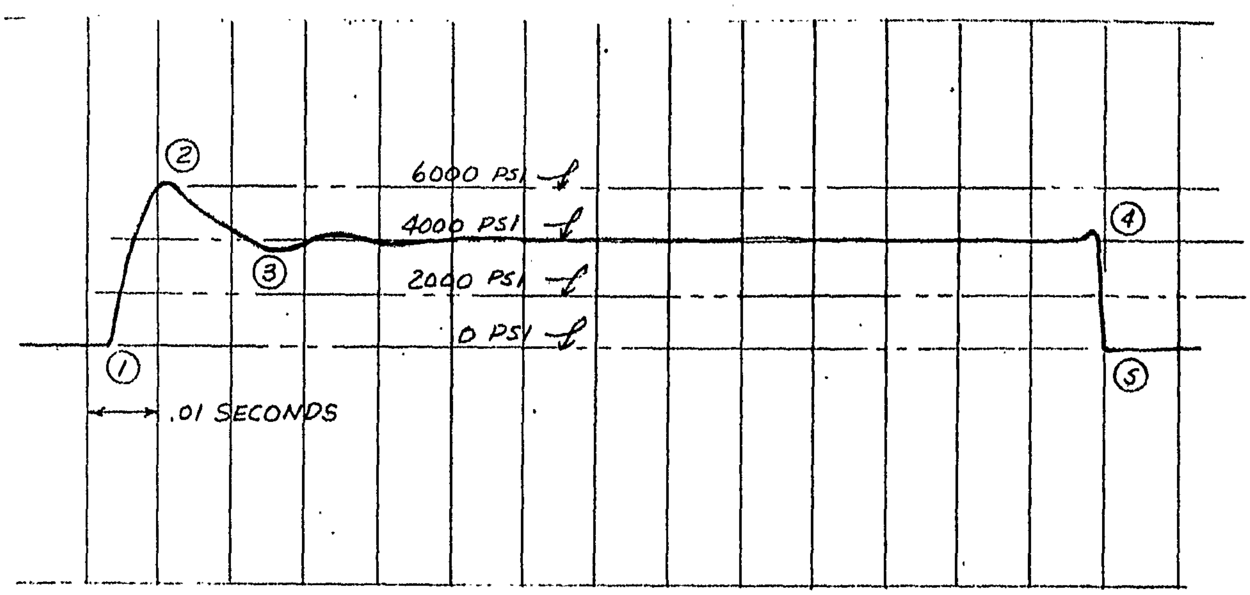


CVA-54049U VIEW OF STATIC METALLIC SEAL IMPULSE TEST SET-UP SHOWING TYPICAL TEST SEALS WITH FITTINGS IN TEMPERATURE BOX.
UNIT 53442 3 JUNE 1959

FIGURE II-22
METALLIC SEAL TEST
OSCILLOGRAPH RECORDING
TYPICAL PRESSURE IMPULSE CYCLE



Cycle sequence + (Ref. Fig. VI-5)

(1) Valves (c) closed (d) open to return

(1) - (2) Valve (c) closed. Valve (d) as shown in Fig. VI-5.
Relief valve (a) controls pressure to intensifier

(2) - (3) Valve (c) opens. Valve (c) remains as shown in Fig. VI-5.
Relief valve (b) controls pressure to intensifier

(3) - (4) Agastat timer delay

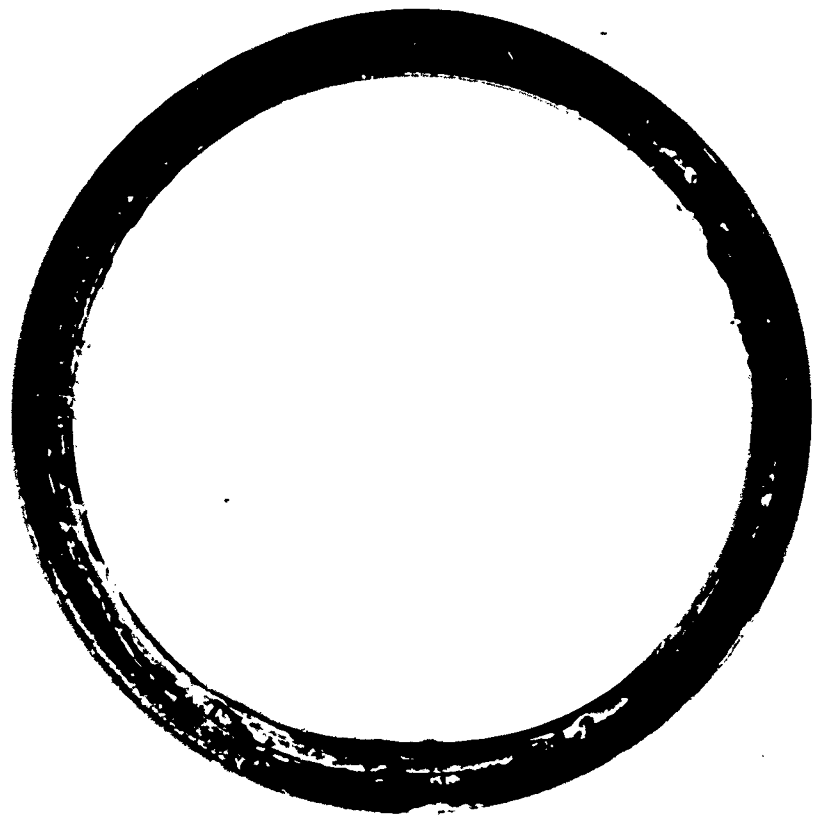
(4) - (5) Valve (c) closes. Valve (d) opens to return.

(5) - (1) Agastat timer delay; ready for next cycle

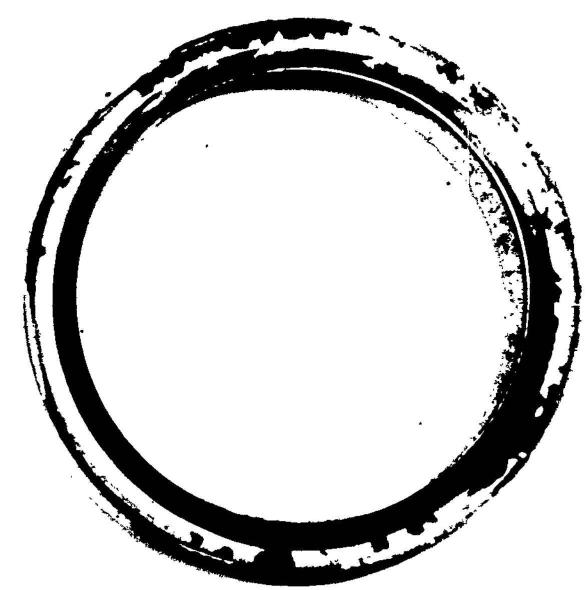
TABLE II-8

HARRISON METALLIC BOSS SEAL TEST

	Manufacturer Part Number	Finish	Total Impulse Cycles	Torque In. Lb. Initial	Torque In. Lb. Final	Leakage
Sample Seals	12100CR 4	Teflon	176,600	100	40	No
Sample Seals	12100CR 6	Teflon	176,600	175	60	Yes
Sample Seals	12100CR 6	Teflon	176,600	175	60	No
Sample Seals	12100CR10	Teflon	176,600	325	600	Yes
Sample Seals	12100CR12	Teflon	176,600	500	800	Yes
	12100CR 6	Teflon	4,600	60	70	Yes
	12100CR 6	Teflon	61,450	300	300	Yes
	12100CR10	Teflon	4,600	132	150	Yes
	12100CR10	Teflon	61,450	684	684	Yes
	12100CR16	Teflon	4,600	237	280	Yes
	12100CR16	Teflon	61,450	1368	1368	Yes
	12100CR20	Teflon	50,000	297	925	Yes
	12100CR20	Teflon	4,600	293	330	Yes
	12100CR20	Teflon	61,450	1692	1692	No
	12100CR28	Teflon	4,600	410	1008	Yes
	12100CR28	Teflon	8,660	2412	3360	Yes
	12100CR28	Teflon	7,090	2412	2412	Yes
	12100CR28	Teflon	0	2412	2412	Yes
	12100AG 6	Gold	4,600	180	180	No
	12100AG 6	Gold	61,450	300	300	No
	12100AG10	Gold	4,600	377	377	No
	12100AG10	Gold	61,450	684	684	No
	12100AG16	Gold	4,600	675	1332	Yes
	12100AG16	Gold	61,450	1368	1368	No
	12100AG20	Gold	50,000	297	1380	Yes
	12100AG20	Gold	4,600	835	1044	Yes
	12100AG20	Gold	61,450	1692	1692	No
	12100AG28	Gold	4,600	1170	1920	Yes
	12100AG28	Gold	61,450	2412	2412	No



VIEW OF HARRISON 12100CR28 TEFLON COATED SEAL AFTER 4,600 PRESSURE IMPULSE CYCLES AND 450°F. UNIT 53442 3 JUNE 1959



VIEW OF HARRISON 12100AG16 GOLD PLATED SEAL AFTER 4,600 PRESSURE IMPULSE CYCLES AND 450°F. UNIT 53442

3 JUNE 1959

TABLE II-9

RECOMMENDED TORQUE VALUES

SEAL SIZE	IN. LB. TORQUE PER NOTE 1	2	3	4
- 4	100 - 150	135 - 150	40 - 57	120 - 153
- 6	175 - 300	270 - 300	60 - 80	180 - 215
-10	400 - 700	650 - 700	132 - 155	377 - 427
-12	500 - 800	900 - 1000	190 - 220	545 - 610
-16		1200 - 1400	237 - 270	675 - 750
-20		1490 - 1705*	293 - 333	835 - 925
-28		2110 - 2440*	410 - 460	1170 - 1260

NOTES:

1. Specification CVC 12-182a - Assembly and Usage of Hydraulic and Pneumatic Hose, Tubing, and Fittings - dated 12 March 1959 (for Flareless Tube Connectors - Steel)

2. AND 10064 Fittings - Installation of Flared Tube, Straight Thread Connectors (AN 818 Nut on Flared Steel Tubing)

3. Harrison Seal Company for Teflon-Coated Seals

4. Harrison Seal Company for Goldplated Seals

5. * Extrapolated Values Based on Seal I.D. Circumference (Inches) to Port Threads per Inch

HARRISON FACE SEAL TEST

A total of 14 Harrison metallic face seals underwent preliminary testing. The face seals tested were of two different configurations - the 12130 configuration (Figure II-23), and the 12110 configuration (Figure II-24). The 12130 configuration is made in sizes up through 0.750 inch outside diameter, while the 12110 configuration is made in sizes of 0.500 inch outside diameter or larger. All seals tested were either goldplated or teflon-coated.

Physical examination, seal installation, static pressure and pressure impulse testing of the seals were performed. Leakage detection during pressure impulse cycling was provided by clear sight tubes partially filled with hydraulic fluid and connected to the non-pressurized test seal cavity. Fluid leaking by the test seal displaced fluid upward in the sight glass which was in full view of the technician operating the test.

Testing of the seals listed in Table II-10 was performed in two groups. In the first group, one seal of each part number, the first listed, was initially set up in manifolds, static pressure tested and then pressure impulse cycled until failure or until a minimum of 50,000 p.i.c. had been performed. The second group consisted of one seal of each part number, the second list, and were tested in the same manner as the first group. The second group of seals were tested in the same manifold cavities as the first; however, any damaged cavity surfaces were first restored by very light sanding with No. 600 grit abrasive paper.

After completion of the normal pressure impulse cycling, the two teflon-coated seals, 12130CR14 S/N 2 and 12110CR150 S/N 1, were set up for reversed pressure testing. The plugs holding the seals in the manifolds were removed and the seals were examined and found to be in good condition. The seals were not removed from the cavity. The plugs were then replaced and static hydraulic pressure was applied. Neither seal passed the static pressure tests, both relieving at pressures below 3,000 psi.

After completion of the pressure impulse cycling, all test manifolds were disassembled for examination of the seal and manifold mating surfaces. In general, the contact surfaces of manifolds and seals showed very little change in all cases where no leakage occurred. In cases of leakage, the seal and manifold mating surfaces usually showed signs of scuffing or erosion. No particular reasons were apparent for variations in leakage for the same size seal. Apparently, leakage which occurred after static pressure tests and during pressure impulse cycling may be attributed to factors not normally detectable, such as minute seal or surface scratches or surface deflections under pressure. The teflon coating on the seals is considered of no practical value under the conditions tested, since in all cases it was found to have peeled away from the sealing surface after being subjected to the 450°F environment and high impulse pressures.

FORM NO. 38

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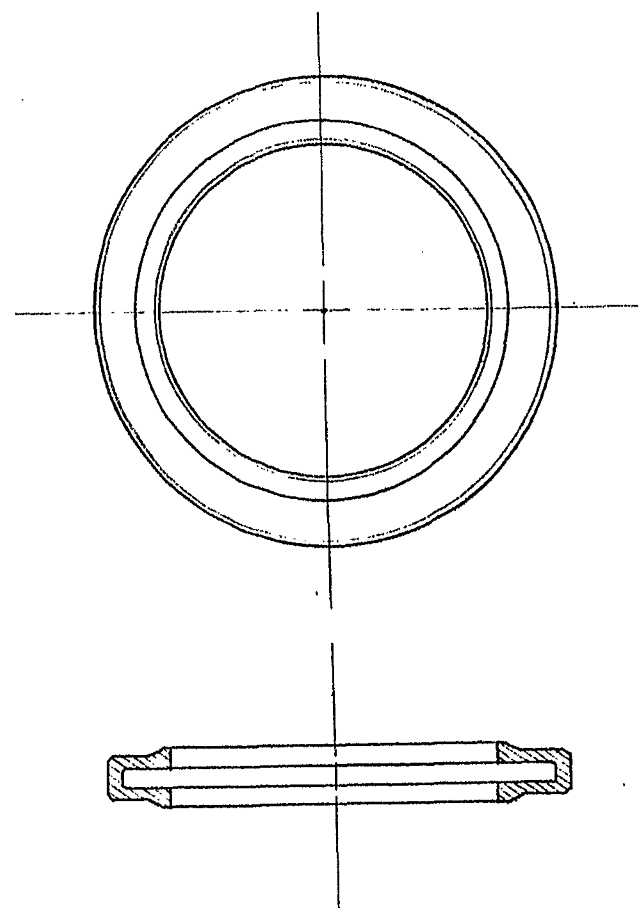
FIGURE II-23

HARRISON MANUFACTURING COMPANY

12130 FACE SEAL



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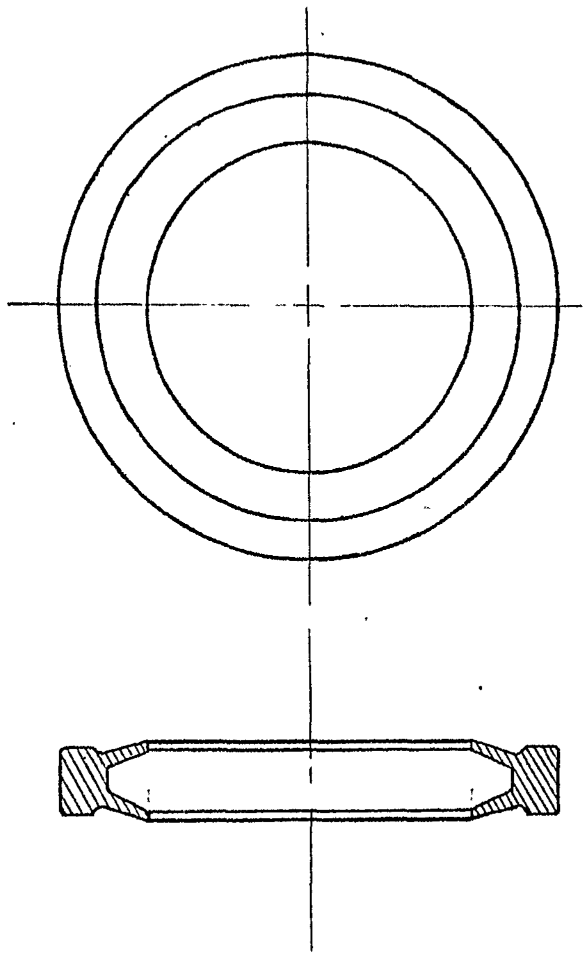
FORM NO. 38

FIGURE II-24

HARRISON MANUFACTURING COMPANY

12110 FACE SEAL



J.J.W
9/23/59

TABLE II-10

HARRISON FACE SEAL TEST DATA

Part Number	Lab S/N	SEAL I.D.(In.)	SEAL O.D.(In.)	SEAL Squeeze (In.)	CAVITY Dia. (In.)	CAVITY Finish (RMS)	Total p.i.c.	Failure
12130CR14	1	.360	.439	.008	.442	16-32	52,000	No
	2	.359	.440	.007	.442	16-32	50,000	No
12130CR22	2	.590	.689	.010	.693	16-32	16,811	Yes
	4	.591	.690	.010	.693	16-32	14,520	Yes
12110CR150	2	1.175	1.496	.011	1.509	16-32	15,104	Yes
	1	1.179	1.496	.011	1.509	16-32	50,000	No
12130PG14	1	.362	.439	.011	.445	16-32	23,879	Yes
	2	.363	.439	.009	.445	16-32	6,650	Yes
12130PG22	2	.592	.689	.008	.692	16-32	12,502	Yes
	4	.593	.689	.009	.692	16-32	19,917	Yes
12110AG125	2	.924	1.246	.012	1.259	16-32	52,000	No
	4	.927	1.246	.012	1.259	16-32	50,000	No
12110AG150	2	1.179	1.496	.012	1.509	16-32	12,502	Yes
	1	1.181	1.498	.013	1.509	16-32	33,890	Yes

NOTES:
1. On part number, CR designates 303 or 304 CRES teflon-coated, AG designates A-286 CRES goldplated, and PG designates 17-4PH CRES goldplated.
2. Finish check made by comparison with a surface check board conforming to ASA-B46, SAE, MIL-STD-10 and NAS 30 standards for designation and control of surface finish of precision machine parts.

Preliminary Hi-Ceal Testing

Preliminary tests were performed on .125 inch and .003 inch cross-section Hi-Ceals early in the program in conjunction with suitability tests being performed on various types of static seals. Extended development effort and testing of the Hi-Ceal is described in Section II and Appendix II-9. Results of the preliminary testing were as follows:

Tests - .125 inch cross-section seals - The first 7/8 O.D. sample seal having a measured depth of 0.1235 inch was installed in a test manifold having a 0.1185 inch cavity depth. A view of the installation is shown in Figure II-25 and the photograph on page 170. The seal was not supported on its O.D. and was required to withstand the full load of the hydraulic pressure. The seal had approximately an 8 RMS finish and the test manifold approximately a 16 RMS finish. Static hydraulic pressures of 10, 4,000 and 6,000 psi were applied to the seal's I.D. without any evidence of leakage. The seal and manifold were then placed in the temperature box and pressure impulse testing was started. The seal had to be removed after 2,690 p.i.c. because of excessive leakage. The manifold was disassembled and the seal and mating surfaces were visually examined. The seal and mating surfaces of the manifold showed considerable erosion, as shown in the photograph on page 171. The manifold hardness ranged from Rockwell C26 to C30.

The finish on the seal was restored by use of No. 500 grit abrasive paper and crocus cloth. The seal was then installed in a manifold capable of applying normal (inside to out) or reversed (outside to in) pressure. The seal after refinishing had a measured depth of 0.1225 inch, and the new cavity depth was 0.1178 inch. Normal static hydraulic pressures to 6,000 psi were applied to the seal without leakage. Reversed pressure was then applied and the seal was found to relieve at 7,750 ± 100 psi. Reversed pressure was applied several times with the same results each time. After relieving, the seal reseated when the pressure dropped below the relieving pressure and sealed again after each pressure application.

Normal hydraulic static pressure was again applied to the seal to substantiate the 7,750 psi as being the seal relief pressure. Pressure was gradually raised until the seal failed at a pressure of 8,500 ± 50 psi. Upon disassembly, the seal was found to have merely unrolled locally on one side without rupture of the metal having occurred. Until the seal failed, no leakage had occurred.

In testing the second seal, hydraulic pressure was always applied in the reverse direction (from O.D. to I.D.). The seal had a measured depth and diameter of 0.8732 - 0.8745 and 0.125 - 0.126 inch, respectively. It was installed in a test manifold having a cavity 0.1130 inch deep by 0.8765 inch I.D. Seal finish and test manifold finishes were approximately 8 RMS and 16 RMS, respectively. Seal wall thickness was 0.019 inch.

The seal passed the static pressure tests satisfactorily and was placed in the temperature box for pressure impulse cycling. After 2,390 p.i.c., the secondary seal (see Figure II-25) began leaking and it had to be replaced. After reassembling the manifold, the test seal was subjected to static pressure tests and was found to be relieving at approximately 4,700 psi. The seal had not leaked during the preceding pressure impulse cycling and it was not removed from the cavity during replacement of the secondary seal. The seal was removed from the manifold and its free depth now measured 0.115 inch, indicating a permanent set of about 0.010 inch had occurred.

The cavity depth was then reduced to 0.1065 inch and the seal was installed in the manifold under its new squeeze of 0.0085 inch (or a total squeeze of 0.0195 inch from the original seal depth). The seal passed the static leakage pressure tests satisfactorily after being subjected to a total of 38,140 p.i.c. without any evidence of leakage. After 44,890 p.i.c., this seal was found to be leaking at room temperature under a static reverse pressure of 1800 ±100 psi. The seal was removed and the free depth measured 0.1085 - 0.1090 inch. The seal was installed again into the same manifold cavity under the same squeeze and a static reverse pressure of 3,500 ± 100 psi was found to cause relieving. The seal and manifold were installed in the controlled temperature box and pressures required to cause leakage at various temperatures were checked. The following conditions were found:

1. Reverse pressure impulse cycled seal at room temperature - seal leaked.

2. Under static reverse pressure with:

    a. Oil temperature of 200°F, seal leaked at 2,500 psi, reseat pressure unknown.
    b. Oil temperature of 300°F, seal leaked at 3,200 psi, reseat pressure at 2,800 psi.
    c. Oil temperature of 400°F, seal leaked at 4,600 psi, reseat pressure at 4,200 psi.
    d. Oil temperature of 450°F, seal leaked at 6,000 psi, reseated at 5,800 psi.

3. Reverse pressure impulse cycled from 0 psi to 6,000 psi when oil temperature reached 450°F. There was no leakage observed.

4. Reverse pressure impulse cycled from 0 psi to 6,000 psi after oil had been at a temperature of 450°F for 30 minutes. There was no leakage observed.

The seal was removed from the test manifold and checked for permanent set. It now had a measured depth of 0.1081 to 0.1085 inch representing a total permanent set of 0.017 inch from the new condition. Visually, the seal appeared unchanged although the faces felt rough.

The third .125 cross-section seal having a measured depth and diameter of 0.1248 - 0.1252 and 0.8750 inch, respectively, was installed in a cavity 0.113 inch deep by 0.8765 inch I.D. Seal finish and test manifold finish was approximately 8 RMS and 16 RMS, respectively. Seal wall thickness was 0.019 inch.

The seal passed the static pressure tests satisfactorily and was placed in the temperature box for pressure impulse cycling in the normal direction. The seal had then been subjected to 50,000 p.i.c. without any evidence of leakage. Upon completing 50,000 p.i.c., the seal test manifold was disassembled and the seal and manifold mating surfaces were visually examined. There was no evidence of erosion or of any scuffing action having occurred, however a very narrow groove (approximately 0.005 inch wide and 0.003 inch deep) had been formed in the test manifold mating surfaces. The manifold hardness ranged from Rockwell C26 to C30.

The seal depth was again measured and found to be 0.117 to 0.119 inch indicating the seal had taken a permanent set of approximately 0.006 inch. The outside diameter now measured 0.8760 to 0.8753 inch.

Tests - .093 cross-section seals - Static hydraulic pressure tests at room temperature were performed on several new seals having various wall thicknesses and cavity dimensions. Three 1.0 inch O.D. seals having a nominal 0.093 inch cross-section diameter and a nominal 0.019 inch wall thickness of 0.086, 0.079, and 0.073 inch depth were installed in cavities without radial support. All three seals were subjected to slowly increasing static hydraulic pressure to 6,000 psi applied internally at room temperature. There was no leakage during pressurization and during a 10-minute period at 6,000 psi. Mating surfaces of the test housings had hardness values ranging from RC 35 to RC 43 and finish values ranging from 6 to 40 RMS.

Two 1.0 inch O.D. seals having a nominal 0.093 inch cross-section diameter and a nominal 0.015 inch wall thickness were installed in cavities of 0.073 of 0.086 inch depth and 1.0 inch inside diameters. Both seals were subjected to slowly increasing static hydraulic pressures to 6,000 psi applied internally, without any leakage occurring. The seals were subjected to slowly increasing static hydraulic pressures in the reversed direction. At 5,950 ± 50 psi the seal in the 0.086 inch cavity collapsed as shown in the photograph on page 172 . The seal installed in the 0.073 inch cavity collapsed under a reverse pressure of 9,000 ± 50 psi. There was no leakage in reverse pressure tests of either seal before collapse. The higher collapse pressure of the seal under the greatest squeeze may be attributed to the greater gripping action imposed upon it by the mating surfaces of the cavity. There was no internal radial support provided for either seal.

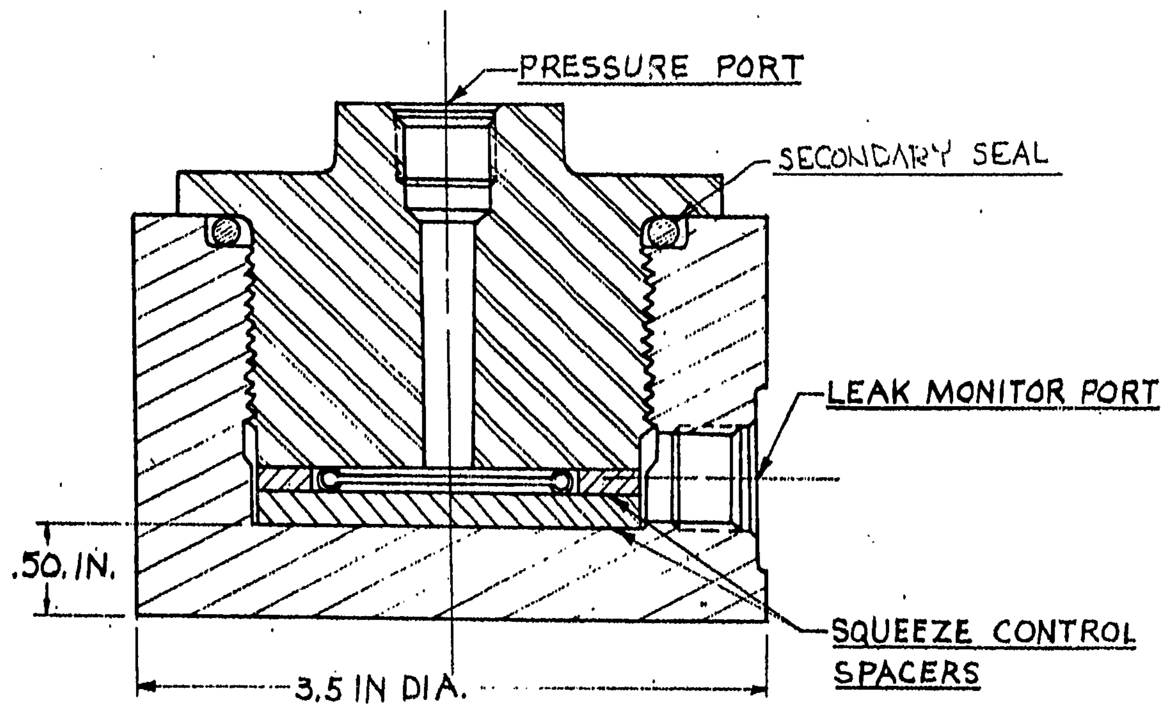
During installation of test seals, unusually high torques were encountered. Torques required for several typical Hi-Ceal installations are shown in Table II-11. Because such torque values would become intolerable in multiple seal installations of a modular unit requiring simultaneous squeezing of all seals, a number of load versus seal deflection tests were run on a compression test machine. Compression tests were run on seals of various wall thicknesses in the annealed and as-formed condition. Results of these tests are shown in Figure II-26. It was noted from the force deflection curves that in both cases of the 0.015 and 0.019 inch-wall seals, the 0.015 inch-wall seal forces were the greater of the two. This was, of course, contrary to what was expected and raised the question as to the seal's uniformity of wall thickness and material condition. Hardness and wall thickness checks were made on a number of seals, the results of which are shown in Table II-12. A wide variance in seal wall thickness and hardness was found. There also exists a wide variation in the exterior dimension of the Hi-Ceal as evidenced by the dimensional check results shown in Table II-13. The seal depth differential existing between the inner and outer rings of the two-way seals would cause a wide variation of squeeze forces on the two parts of the seal.

To determine what portion ofthe high torque values may be attributed to thread friction, a test was set up to measure the torque required to cause rotation of a threaded plug under various axial loads. The threaded plug and housing were physically identical to and of the same material as that used in measuring installation torques on the Hi-Ceals. (See Section III, Figure III-28.) As shown in Table II-11, a torque of 859 in./lb. was required to deflect a 1-3/8 inch O.D. seal 0.008 inch. The force required to deflect a similar 1-3/8 inch

O.D. seal the same amount in a straight compression test was 1,560 lbs., per Figure II-27. The torque required to rotate the same plug under a 1,560 lb. axial load would be 690 in.-lb., per Figure III-28. Although there is a possibility of considerable difference of deflection loads between the two seals, it is obvious that thread friction accounts for the greater portion of the seal installation torque.

FIGURE II-25

HI-CEAL IN TEST MANIFOLD



CVA HOUSING

(MATERIAL - ALL CRES. STEEL)

[illegible] in [illegible]

[illegible] No. [illegible] manifold showing [illegible] by [illegible] Pressure Impulse [illegible]

- A VIEW OF 1.0 INCH O.D. X .093 INCH CROSS SECTION DIAMETER X .015 INCH WALL THICKNESS HI-SEAL AFTER COLLAPSE AT 5950±50 PSI DURING APPLICATION OF REVERSE PRESSURE

TABLE II-11

TORQUE ON 2-1/4 - 12 UNTHREADED PLUG TO DEFLECT HI-SEAL

SEAL S/N	SEAL (Inches Nominal)			APPLIED TORQUE (In./Lb.)	SEAL DEFLECTION (Inches)
	Outside Diameter	Cross-Section	Wall Thickness		
17	2.750	.093	.015	1,602	.0044
18	2.750	.093	.015	3,240	.0064
19	2.750	.093	.015	3,840	.0069
21	1.375	.093	.015	859	.0080
22	1.375	.093	.015	1,320	.0160

NOTES:

1. Seal, seal cavity and threads were lubricated with MLO-8200 hydraulic fluid before installation.

2. Seal material was Type 304 stainless steel and torques were measured on seals in the as-formed condition.

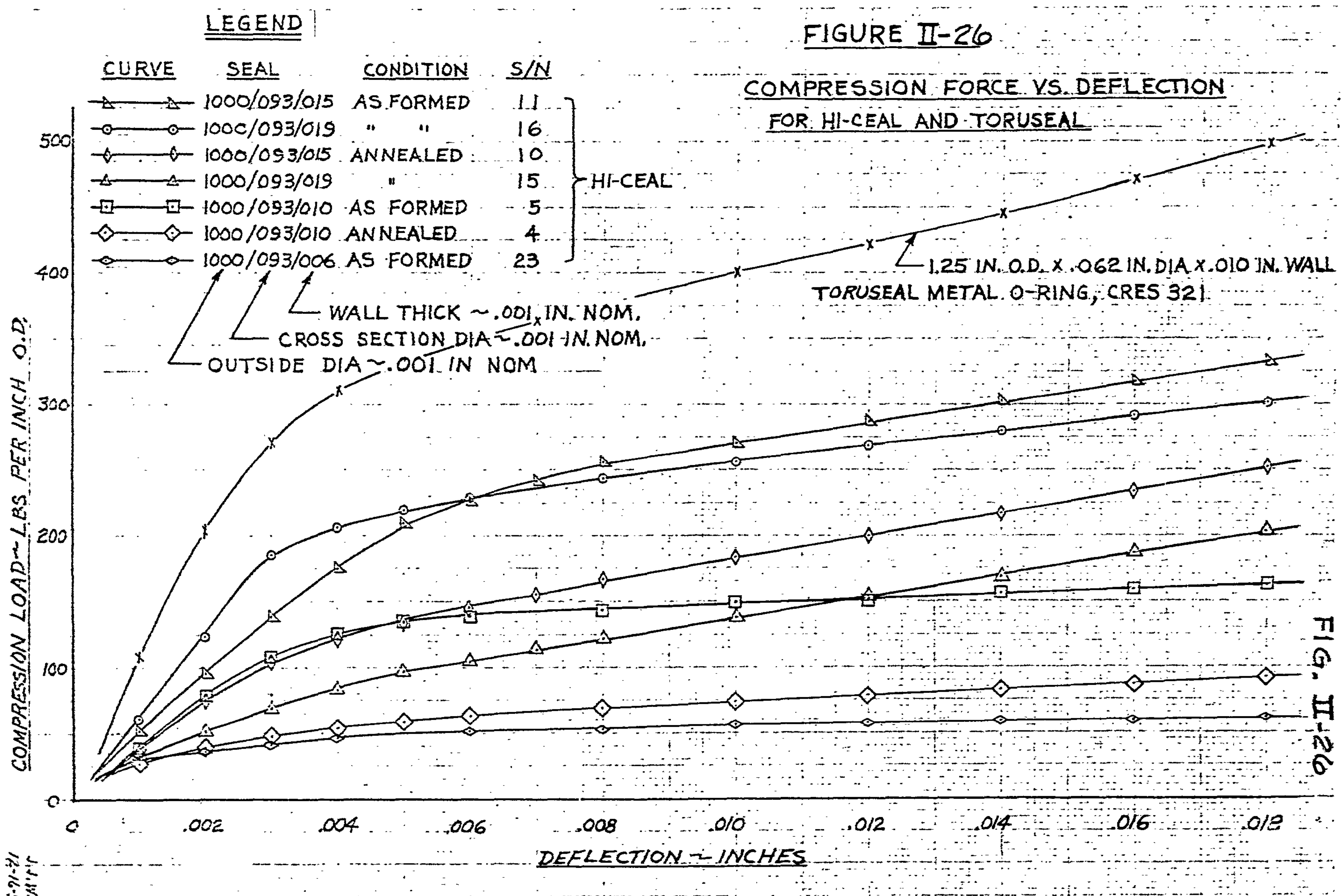


FIGURE II-26
COMPRESSION FORCE VS. DEFLECTION
FOR HI-CEAL AND TORUSEAL
LEGEND
CURVE SEAL CONDITION S/N
1000/093/015 AS FORMED 11
1000/093/019 " " 16
1000/093/015 ANNEALED 10
1000/093/019 " 15
1000/093/010 AS FORMED 5
1000/093/010 ANNEALED 4
1000/093/006 AS FORMED 23
HI-CEAL
WALL THICK ~.001 IN. NOM.
CROSS SECTION DIA ~.001 IN. NOM.
OUTSIDE DIA ~.001 IN NOM
1.25 IN. O.D. X .062 IN. DIA X .010 IN. WALL
TORUSEAL METAL O-RING, CRES 321
COMPRESSION LOAD ~ LBS PER INCH O.D.
DEFLECTION ~ INCHES
0
100
200
300
400
500
.002
.004
.006
.008
.010
.012
.014
.016
.018
J.J.W.
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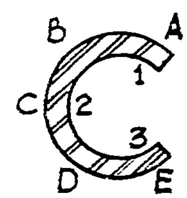
FIG. II-26

HI-CEAL MATERIAL HARDNESS AND WALL THICKNESS TEST RESULTS

AREA		SEAL S/N 4		SEAL S/N 5		SEAL S/N 15		SEAL S/N 16	
WALL (See Sketch)	HARDNESS	WALL THICKNESS	HARDNESS	WALL THICKNESS	HARDNESS	WALL THICKNESS	HARDNESS	WALL THICKNESS	HARDNESS
1	A	.0145 (In)	RB 78.5	.0125 (In)	RC 21.0	.0180 (In)	RC 15.5	.0160 (In)	RC 21.0
2	B	.0160	RC 16.0	.0150	RC 33.0	.0170	RC 1.0	.0160	RC 23.0
3	C	.0140	RC 24.0	.0120	RC 26.0	.0170	RC 10.0	.0160	RC 20.0
1	D	.0130	RC 23.0	.0130	RC 25.5	.0180	RC 3.0	.0170	RC 18.0
2	E	.0150	RC 3.0	.0160	RC 19.0	.0170	RC 1.0	.0170	RC 15.0
3		.0120		.0135		.0185		.0170	
1						.0180		.0165	
2						.0170		.0190	
3						.0190		.0160	



NOTES:

1. Areas where hardness and wall thickness measurements were made are shown in sketch. Measurements were repeated at various cross-sections of same seal.
2. Nominal wall thickness of S/N 4 and 5 seals was 0.010 inch, and was 0.019 inch for S/N 15 and 16.
3. Wall thickness measurements shown are in inches.

TABLE II-12

TABLE II-13

HI-CEAL DIMENSIONAL CHECK RESULTS

Part Number	Type	S/N	A Maximum	A Minimum	B-INNER Maximum	B-INNER Minimum	B-OUTER Maximum	B-OUTER Minimum
1000/093/10/A/0	1-way	4	1.0003	.9985	.0940	.0930	-	-
	1-way	5	1.0002	.9990	.0930	.0930	-	-
1000/093/15/A/0	1-way	6	1.0003	1.0001	.0930	.0922	-	-
	1-way	7	1.0003	.8998	.0926	.0905	-	-
1000/093/10/A/0	2-way	8	1.1504	1.1498	.0940	.0929	.0932	.0896
	2-way	9	1.1511	1.1495	.0940	.0931	.0934	.0930
1000/093/15/A/0	2-way	10	1.1529	1.1524	.0926	.0915	.0935	.0929
	2-way	11	1.1542	1.1512	.0926	.0905	.0930	.0932
1000/093/19/A/0	1-way	12	1.0200	1.0170	.0937	.0926	-	-
	1-way	13	1.0184	1.0170	.0936	.0929	-	-
	1-way	14	1.0182	1.0167	.0935	.0924	-	-
2750/093	1-way	7	2.7667	2.7490	.0940	.0929	-	-
	1-way	8	2.7604	2.7455	.0930	.0924	-	-
	1-way	19	2.7560	2.7544	.0940	.0932	-	-
	1-way	20	-	-	.0934	.0918	-	-

NOTES: 1. Part number code is as follows:

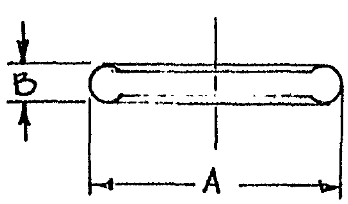
1000/093/10/A/0

Seal nominal wall thickness - 0.001 inch nominal

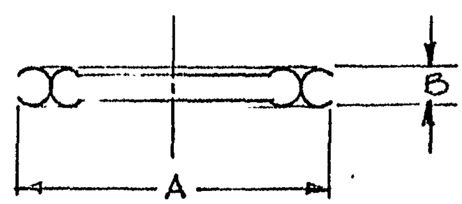
Seal nominal depth - 0.001 inch nominal

Seal nominal diameter - 0.001 inch nominal

2. Table headings have following code:

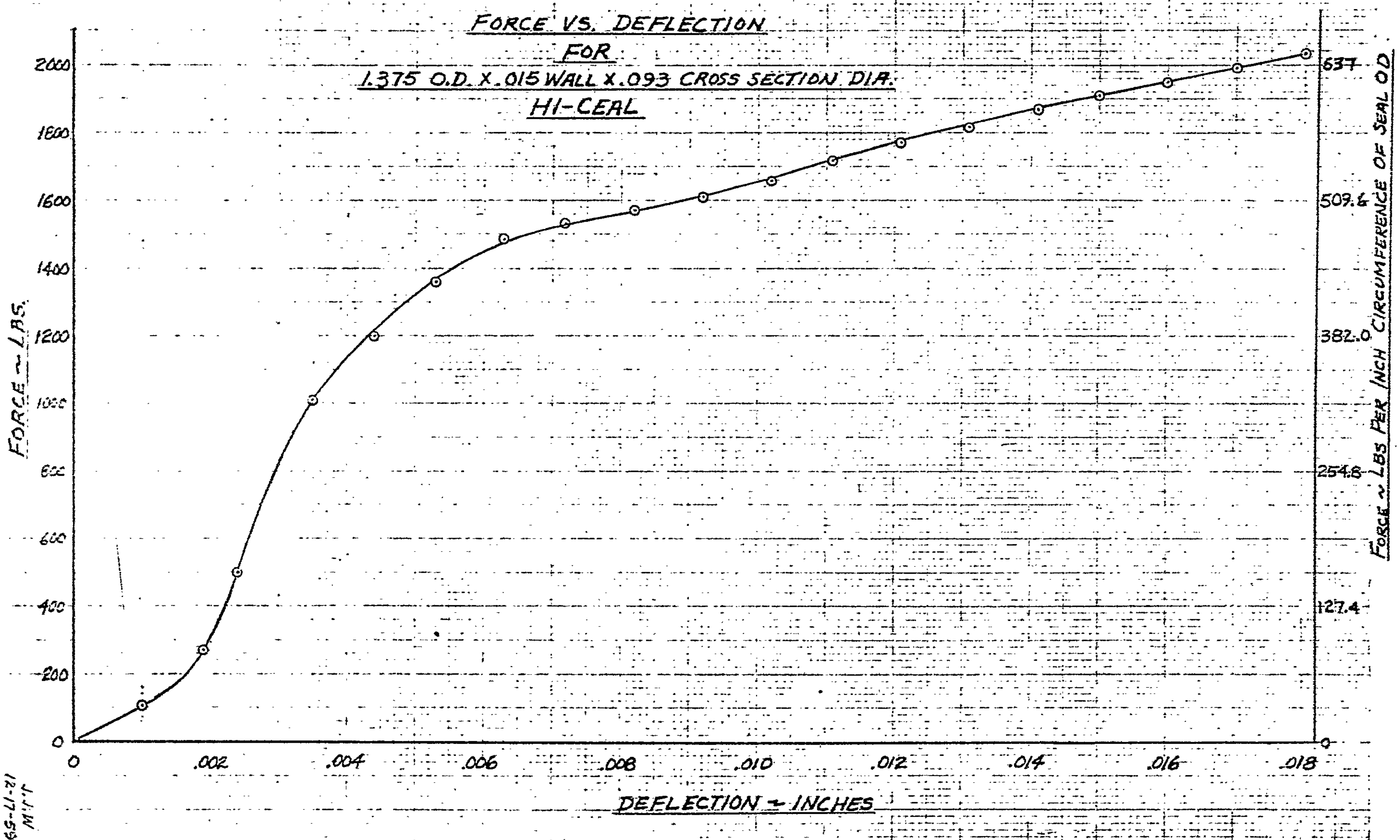


1-WAY SEAL



2-WAY SEAL

FIGURE II-27


FORCE VS. DEFLECTION
FOR
1.375 O.D. X .015 WALL X .093 CROSS SECTION DIA.
HI-CEAL
FORCE ~ LBS.
0
200
400
600
800
1000
1200
1400
1600
1800
2000
0
.002
.004
.006
.008
.010
.012
.014
.016
.018
DEFLECTION ~ INCHES
FORCE ~ LBS PER INCH CIRCUMFERENCE OF SEAL OD
127.4
254.8
382.0
509.6
637
J.J.W
12-17-59

NAVAN SEAL TEST

A total of eleven NAVAN metallic boss seals (Figure II-28) underwent testing. Seal inspection, installation and test procedures were similar to that used in the Harrison seal tests. The same equipment was used for the pressure impulse testing. Torque values as recommended by the manufacturer for aluminum seals on aluminum or carbon steel fittings are shown in Table II-14. The manufacturer made no specific recommendations as to torque required for stainless steel seals on stainless steel fittings in all sizes. It was stated, however, that 20 to 30 ft./lb. additional torque was required for the -6 and -8 size stainless steel seals on stainless steel fittings.

Two each of a -6, -10, -16, -20 and -28 size seal were installed on fittings and torqued to the values shown in Table II-14. Seals were identified by S/N 1 and 2 for each size. All seals were static pressure tested. Torque on both -28 seals had to be increased to 360 ft./lb., -20 S/N 2 to 270 ft./lb., and -10 S/N 2 to 135 ft./lb. before passing the static leakage test satisfactorily. Finish on all fitting mating surfaces was compared with a standard surface finish board and was considered better than 32 RMS.

The -6, -10 and -16 size seals were subjected to 61,163 p.i.c. without any evidence of leakage. After 11,809 p.i.c., the -20 and -28 size seals were leaking. It was necessary to torque the -28 seals to over 500 ft./lb. to stop leakage. The -20 seals were removed and replaced with Harrison seals, one teflon-coated and one goldplated, and each identified by S/N 1. The replacement was made because the NAVAN seal torque was already considered excessive and this was a good opportunity to compare the performance of another seal installed on the same fittings. The result of this test is given in the Harrison test results section.

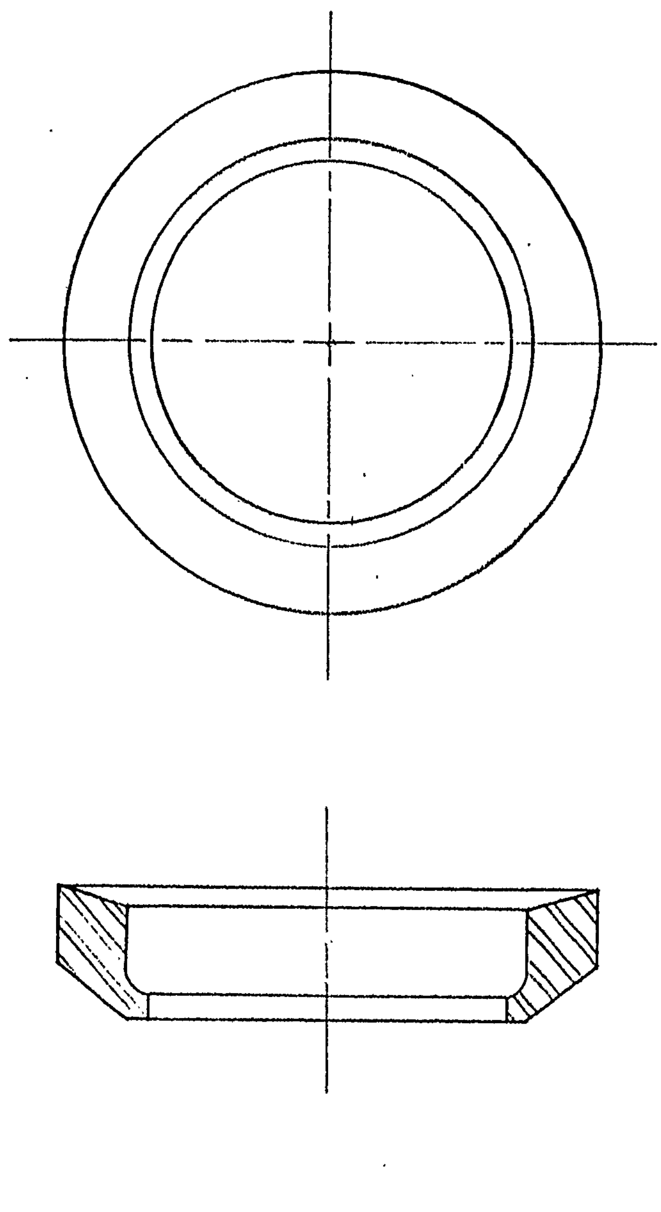
In completing the 61,163 p.i.c., the torque on the -28 seals ultimately was raised to over 1,000 ft./lb. to prevent leakage.

A -28 S/N 3 NAVAN seal installed in another manifold and torqued to 375 ft./lb. would not pass the static leakage test. It was necessary to raise the torque to 500 ft./lb. before effecting a seal. The seal has since been subjected to 12,841 p.i.c. without evidence of leakage.

FORM NO. 38

FIGURE II-28

NAVAN PRODUCTS INC. SEAL FOR AND10050 BOSS



J. J. W
6/22/43

TABLE II-14

NAVAN SEAL TORQUE VALUES

Seal Size	Ft./Lb. Torque Per note 1	2	3	Lab S/N
- 6	30 - 35	52 52	52 52	1 2
-10	115 - 130	75 75	75 135	1 2
-16	165 - 175	165 165	165 165	1 2
-20	195 - 205	200 200	200 270	1 2
-28	230 - 250	238 238 375	360 360 500	1 2 3

NOTES:

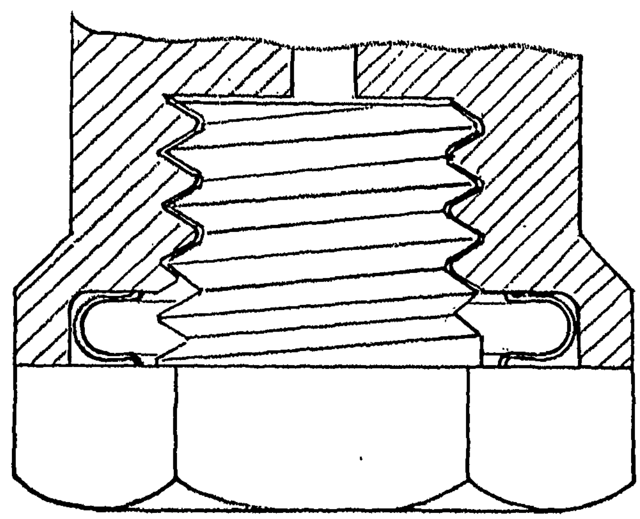
1. Vendor's recommendation
2. Initial installation
3. Highest value used to effect seal under test condition.

OMEGA SEAL TEST

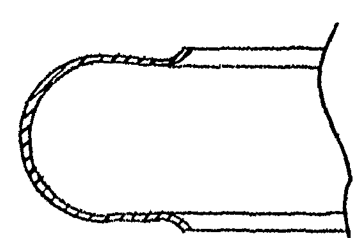
One Omega seal already installed in a manifold was obtained as a static seal test sample. A detail of the seal and its installation is given in Figure II-29. Since these seals are not available for production in the larger sizes, no special testing other than impulse testing was performed. The manifold was connected to the pressure impulse cycling set-up and impulsed whenever other seals were being tested. To date, the seal has been cycled 131,000 times with no leakage. It should be noted, however, that the installation is such that the full impulse may not reach the seal since the pressure wave must first pass through the threaded portion of the manifold. This can account for the long, trouble-free life of the seal.

FIGURE II-29

OMEGA SEAL



TYPICAL INSTALLATION



SEAL CROSS SECTION

RACO SEAL TEST

One test sample of the Raco seal was received for preliminary investigation as a possible face seal.

The Raco seal consists of a teflon cover molded over a steel inner skeleton as shown in Figure II-30. A test manifold that can be adjusted to various degrees of squeeze for the test seal was fabricated. The seal was installed in the housing for testing as shown in Figure II-31. Initially, the seal was installed with a cavity depth of 0.200 inch which imposed a 0.046 inch squeeze on the seal. With this squeeze, the seal would not pass the static pressure test. Cavity depth was then reduced to 0.186 inch, which included .006 inch clearance between the housing and plug. The purpose of the 0.006 inch clearance was to simulate the approximate standard module radial clearance for that size seal. Static hydraulic pressure of 10, 4,000 and 6,000 psi was applied to the seal without evidence of failure or leakage.

The seal and test housing were then installed in the controlled temperature box and the addition of heat and pressure impulse cycling was begun simultaneously. After 667 p.i.c.,the seal blew out as the oil temperature reached 380°F. Failure occurred as a result of the softening and extrusion of the seal's teflon cover. A view of the seal taken after failure is shown in the photograph on page 186.

FORM NO. 38

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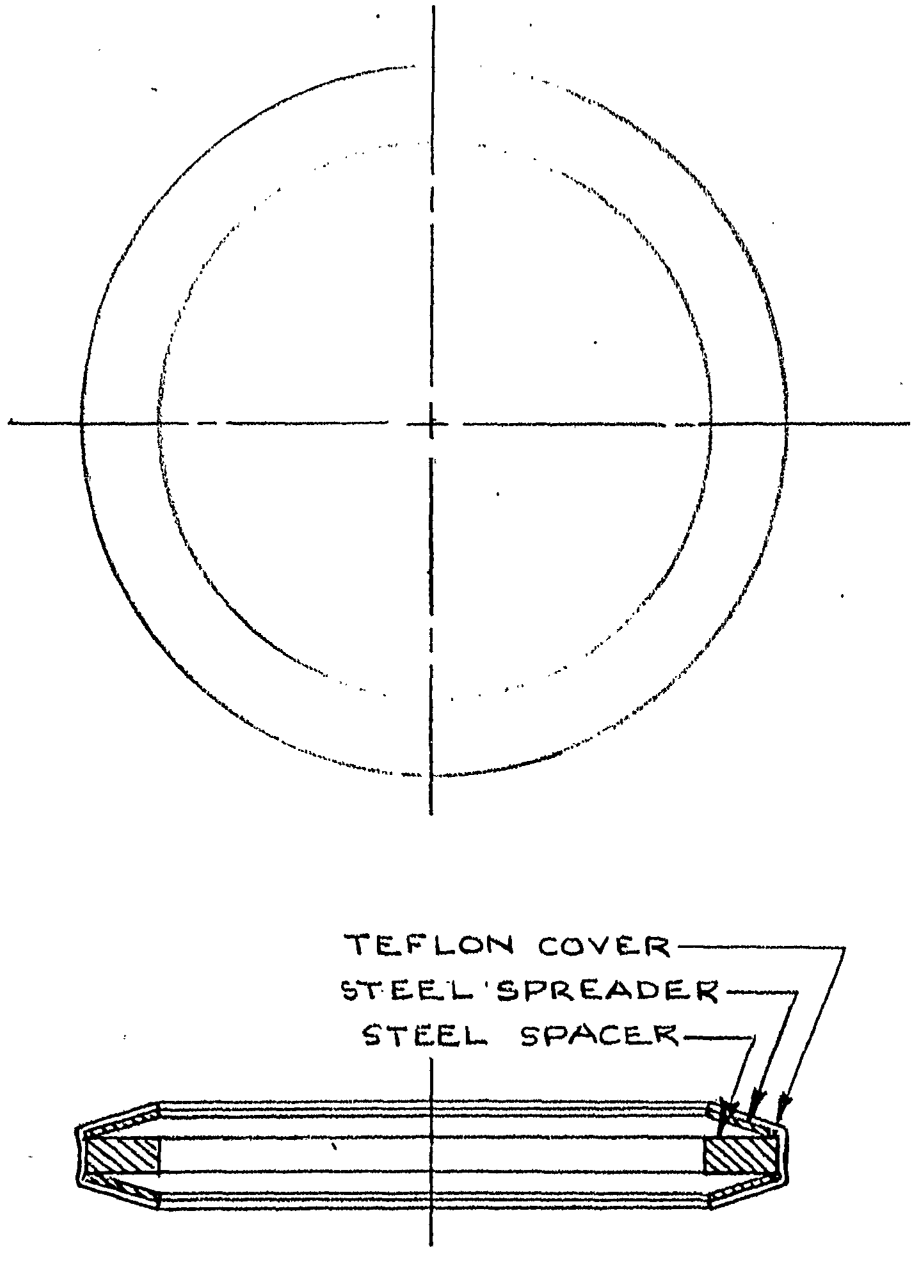
FIGURE II-30

RACO SEAL



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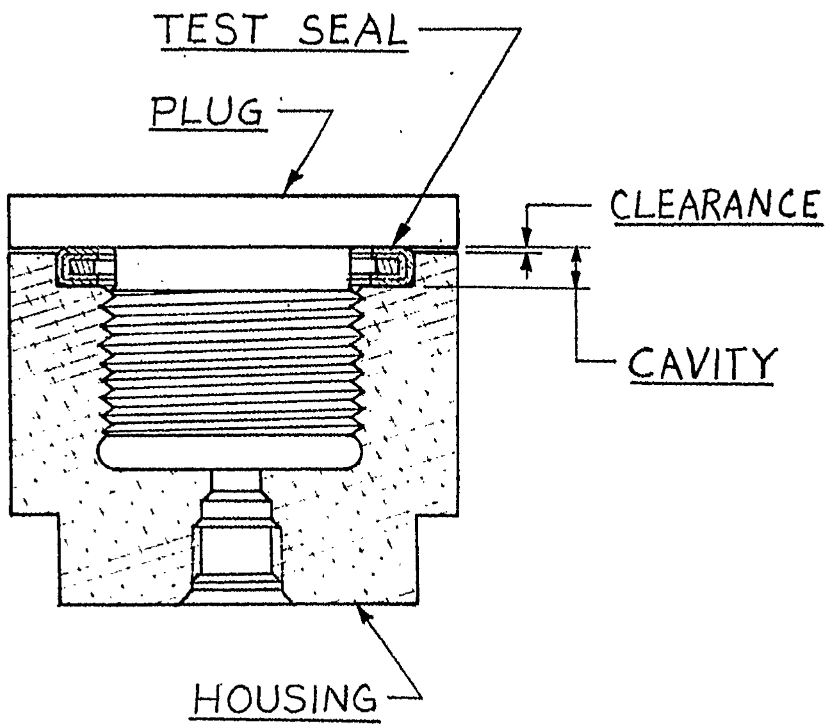
6-1-9

FIGURE II-31

RACO SEAL IN TEST HOUSING



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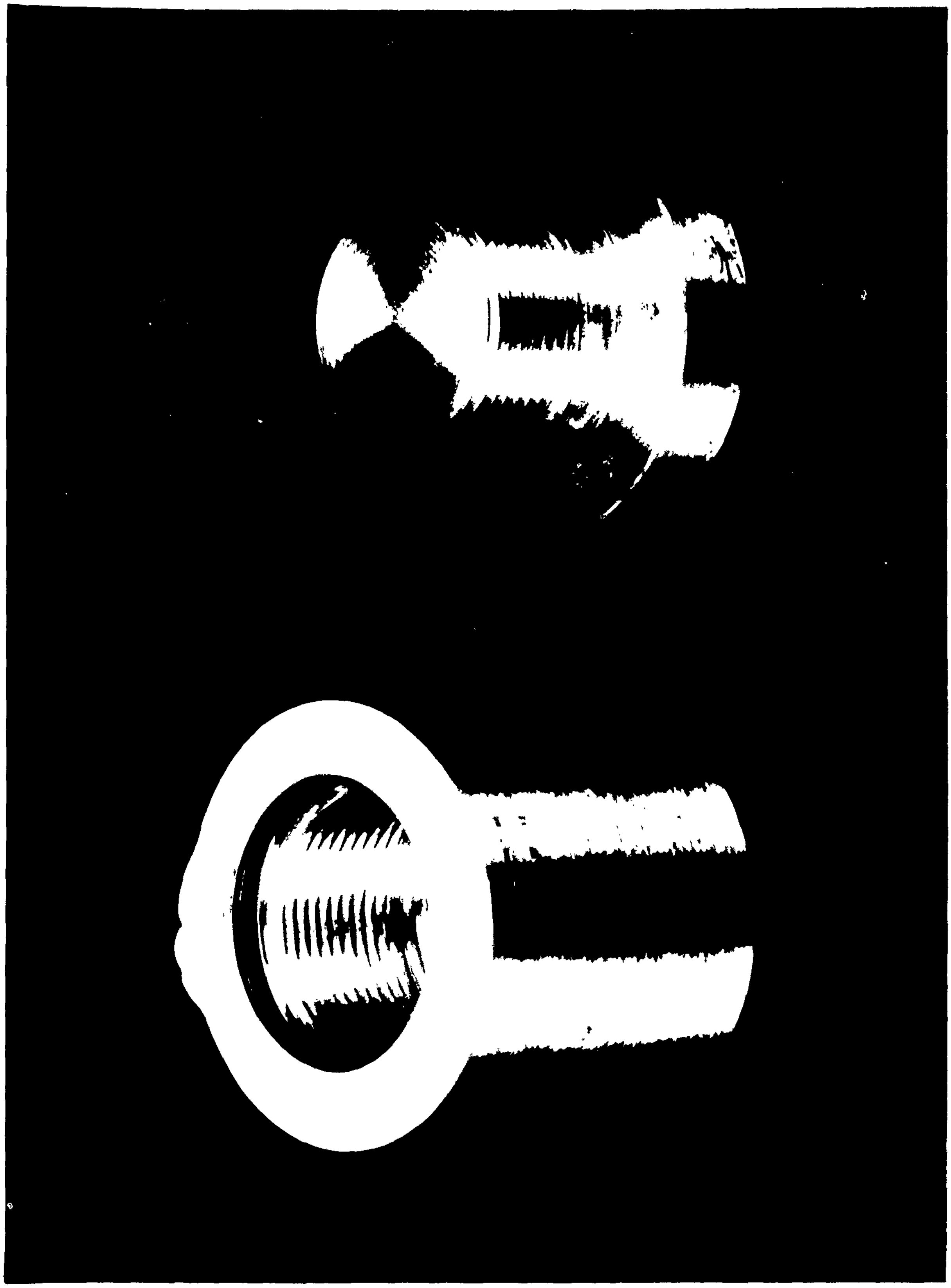


Modular Hydraulics - Failed Saco Seal in Test Manifold

TORUSEAL TEST

Tests were completed on 8 toruseals manufactured by the D.S.D. Manufacturing Company of Hamden, Connecticut. Toruseals are hollow metallic O-rings, the construction of which is similar to the metallic O-rings previously tested. The significant difference in the toruseal compared to metallic O-rings previously tested is in the cross-section diameter of the tubing of the seal. Nominal cross-section diameter of the tubing of the toruseals tested was 0.062 inch compared to 0.035 inch previously tested metallic O-rings.

Physical inspection, seal installation, static pressure and pressure impulse cycling tests were performed. Leakage detection during pressure impulse cycling was provided by clear sight glasses partially filled with fluid and connected to the non-pressurized test seal cavity. Fluid leaking past the test seal displaced fluid upward in the sight glass in full view of the technician operating the test.

The results of the tests along with physical data for the seals and cavities in which they were installed are shown in Table II-15. The goal for the dynamic tests was to impose 50,000 normal p.i.c. followed by 25,000 reversed p.i.c. Only those seals completing the normal pressure tests were tested in the reversed direction.

In each seal installation two seals were required, the test seal and one secondary seal which was physically the same as the test seal except that it was of a larger diameter. The secondary seal was necessary to enable the application of reversed pressure to the test seal, i.e., the building up of pressure on the outside diameter of the test seal. The force required to squeeze the two seals simultaneously during installation was found to be objectionably high although the seals, cavities and plug threads were lubricated with the test fluid, MLO-8200. Torque to install two seals simultaneously ranged from 750 in./lb. on a 1-UNF thread for 3/8 in. O.D. and 5/8 in. O.D. seals to 3,500 in./lb. on a 2-12 UN thread for 1-1/4 in. O.D. and a 1-3/4 in. O.D. seals. Compression forces required to squeeze a 1.25 in. O.D. seal were measured on a compression test machine. Forces required per inch of seal circumference are shown in Figure II-26.

TABLE II-15

TORUSEAL FACE SEAL TEST DATA

Part Number(1)	Lab S/N	SEAL (In.±.001) O.D.	SEAL (In.±.001) Depth	CAVITY (In. ±.001) O.D.	CAVITY (In. ±.001) Depth	Cavity Finish RMS (2)	Pressure Impulse Cycles (3) Normal	Pressure Impulse Cycles (3) Reverse	Remarks
A-375-C-TC	1	.381	.064	.381	.048	32-40	50,000	-	No leaks or failure. Secondary seal leaked during reverse proof pressure
A-625-C-TC	1	.630	.064	.631	.044	20-32	50,000	4,063	No leaks or failure. Removed due to secondary seal leakage.
A-1000-C-TC	1	1.009	.067	1.005	.044	20-32	16,870	0	Seal began leaking.
A-1250-C-TC	1	1.251	.062	1.257	.046	25-32	16,870	0	Seal began leaking.
A-375-C-AG	1	.377	.062	.380	.045	25-32	50,000	25,000	No leaks or failure.
A-625-C-AG	1	.628	.061	.629	.044	20-32	50,000	25,000	No leaks or failure.
A-1000-C-AG	1	1.005	.054	1.004	.046	20-25	50,000	17,737	No leaks or failure. Removed due to secondary seal leakage.
A-1250-C-AG	1	1.250	.064	1.254	.045	45-64	16,870	0	Seal began leaking.

NOTES: (1) Seals manufactured from 0.062 inch diameter x 0.010 inch wall thickness type 321 stainless steel, TC designating teflon-coated and AG designating silverplated.

(2) Finish check made by comparison with a surface check board conforming to ASA-B46, SAE, MIL-STD-10 and NAS 30 standards for designation and control of surface finish of precision machine parts.

(3) Test target was to impose 50,000 normal p.i.c. plus 2,500 reversed p.i.c. Only those seals completing the normal p.i.c. were set up for reversed p.i.c.

WADC SEAL TEST

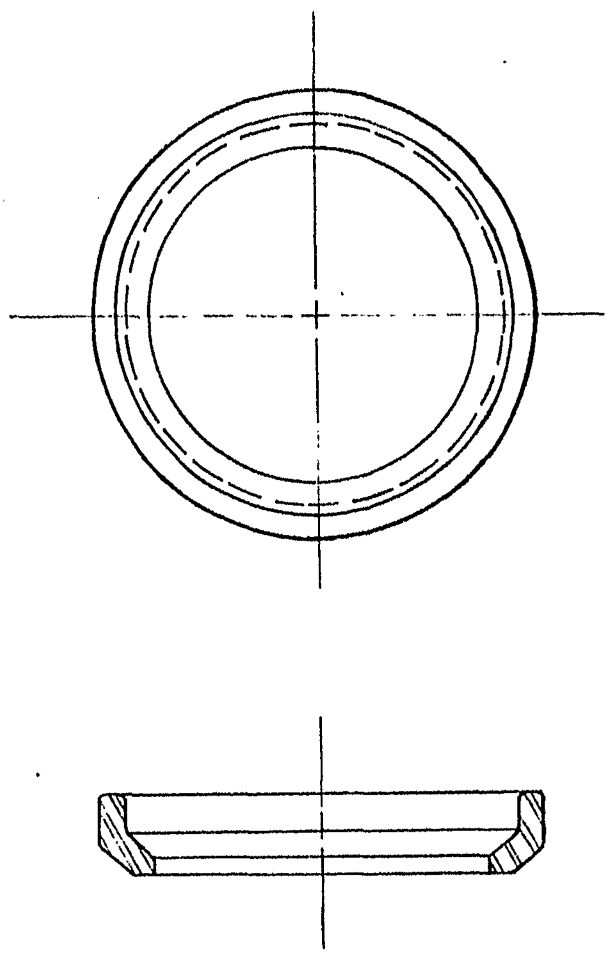
Eight metallic boss seals of the WADC 101 design underwent testing. Seal inspection, installation and test procedures were similar to those used in the Harrison seal tests. The same equipment was used for the pressure impulse tests. Seals tested were two each of a -6, -10, -16 and -20 size. Seals were identified by S/N 1 and 2 in each size. The two -6 and -10 S/N 2 seals were nickelplated while the remaining seals had no special coating or finish.

All seals were installed on fittings, preset, lubricated and torqued to the AND 10064 recommended torque values. All seals passed the static leakage test satisfactorily after the torque was increased to 2,400 in./lb. Seals and fittings were then connected to equipment for pressure impulse testing. A total of 59,000 p.i.c. were applied to all seals. After 27,820 p.i.c., one -16 seal began leaking. Torque was increased to 2,880 in./lb. without effecting a seal. A seal was effected after raising the torque to 3,280 in./lb. No further leakage occurred with this particular seal. The other -16 seal and the -20 seal began leaking after 47,217 p.i.c. Torque was raised to 1,864 in./lb. on each seal and leakage stopped. No other leakage or failure occurred throughout the test.

Two disadvantages of the WADC seal are the fact that it requires a pre-setting tool and torque required to effect a seal is high. No additional tests are planned for the WADC seal. Seal configuration is shown by Figure II-32.

FIGURE II-32

WADC METALLIC SEAL
FOR AND-10050 PORT



J. J. W.
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APPENDIX II-5

GENERAL TEST PROCEDURES FOR METALLIC SEALS

CVC 2461 - RADIAL SEAL

CVC 2464 - FACE SEAL

CHANCE VOUGHT CORPORATION
VOUGHT AERONAUTICS DIVISION

ENGINEERING DEPARTMENT SPECIFICATION

Originated
By: J. M. Spell

CVC 2461

PROCUREMENT SPECIFICATION

FOR

METALLIC RADIAL SEALS

NOTE: The last page of this specification will be Dwg. CVP-3352.

CHANCE VOUGHT CORPORATION CVA 2461

ENGINEERING DEPARTMENT SPECIFICATION

1. SCOPE. - This specification states the design, manufacturing, and testing requirements for metallic radial seals to be used for static sealing applications in a 4000 psi, 450°F hydraulic system. This specification further outlines the procedure for development of the seal desired.

2. DESIGN AND MANUFACTURING REQUIREMENTS. -

2.1 General. - Seals and seal installations shall be compatible with the fluid, thermal, and functional requirements specified herein. ll seals and related parts shall be metallic.

2.2 Materials. - Materials shall conform to applicable government specifications. Materials which are not covered by applicable specifications may be used provided it can be demonstrated that their use will result in a superior product.

2.3 Corrosion Protection. - Metals shall possess adequate corrosion-resistant characteristics or shall be suitably protected by the use of coatings which conform to applicable government specifications. Other metallic or non-metallic coatings which can be demonstrated to provide adequate protection against corrosion may be used provided their use is compatible with the fluid, thermal, and functional requirements specified herein.

2.4 Fluid. - The hydraulic fluid used shall be MLO-8200 which may be purchased from the Oronite Chemical Company, 200 Bush Street, San Francisco, California.

2.5 Temperature Range. - The seal shall be capable of full performance with the fluid at any temperature throughout the range of -65°F to 450°F and shall seal satisfactorily within a range of ambient temperatures from -65°F to 650°F.

2.6 Rated Pressure. - The rated hydraulic pressure shall be 4000 psi.

2.7 Re-Usability Versus Cost. - It is desirable that seals be of such a design as to permit their re-use a minimum of two times after operating pressure has been applied.

2.8 Machining Tolerances and Surface Finishes. - The seals must function on sealing surfaces where finish may vary between zero and 16 RHA. (RHA is defined in MIL-STD-10.) It is desirable that machining tolerances on the valve and on the cacity sealing surfaces (see Figure III) be a generous as possible. The seal manufacturer shall establish these tolerances based on test results of Phase II.

2.9 Removability. - The seal design should be such that deformation of the seal against the cavity bore does not occur so as to cause difficulty or require special tools for normal installation and removal of the seal.

2.10 Lubricants. - No lubricant should be required for seal installation or performance other than hydraulic fluid MLO-8200.

2.11 Seal Tightening Forces. - It is desirable that the installation torque values required for effective sealing be held as low as possible. In no case, however, shall the torque required to effect zero-leakage exceed 100 ft-lbs.

3. TEST REQUIREMENTS. -

3.1 Proof Pressure. - Each seal shall be subjected to proof pressure of 6000 psi at 450°F with pressure applied to one side of the seal for 5 minutes and then applied to the other side of the seal for 5 minutes. There shall be no leakage and no permanent deformation or other visible damage to the seal.

3.2 Burst Pressure. - The seal shall withstand a burst pressure of 10,000 psi at room temperature. Burst pressure shall be applied for 5 minutes against one side of the seal and then applied for 5 minutes against the other side of the seal. There shall be no leakage by the seal in either direction. The burst pressure test shall be the last test performed.

3.3 I[illegible]e Test. - The pressure impulse tests shall be conducted at a rate of 35 ± 5 cp[illegible]ch impulse cycle shall constitute a rise from 0 psi to surge pressure and drop [illegible] zero as shown in Figure I. Hydraulic fluid shall be used as a test medium with a peak surge pressure of 1.43 to 1.57 times the rated pressure as shown by an oscilliscope. Pressure impulses shall be applied alternately to one side of the seal and then applied to the other side. The seal shall have 100,000 impulse cycles alternately applied to each side of the seal for a total of 200,000 pressure impulse cycles. The impulse cycles shall be imposed in accordance with paragraph 3.4 below. Before and after the impulse tests, the seals shall be proof pressure tested per paragraph 3.1. There shall be no leakage during the impulse tests or proof pressure tests.

3.4 Temperature-Time Spectrum. - The impulse test (200,000 cycles) shall be conducted while the seal undergoes the temperature-time spectrum shown on Figure II. Each spectrum should take approximately 6 hours to complete and will consist of 11,120 cycles. Eighteen days are required for the test. The first, third, fifth, tenth, thirteenth, and sixteenth spectrums shall begin after the test set-up has soaked at -65°F for 8 hours (over night). The remaining spectrums shall begin at 95 ± 25°F on the second, fourth, sixth, seventh, eighth, ninth, eleventh, twelfth, fourteenth, fifteenth, seventeenth and eighteenth days of the test. The rate of temperature rise shall be within the shaded areas shown on Figure II.

4. SEAL TYPE, SIZES AND CAVITY CONFIGURATION. -

4.1 General. - The metallic radial seal covered by this specification is defined as an assembly of paired, approximately wedge-shaped, sealing rings to be "loaded" by mechanical force against the circumference of a cartridge-type unit inserted in a manifold and correspondingly loaded against the inside diameter of

the bore into which the cartridge unit is inserted. Tubular spacers between several pairs of sealing rings are integral parts of the seal assembly and serve to locate the seals as well as to act as load-carrying members between seals.

4.2 Cavity Configuration. - The drawing attached to this specification as Figure III shows the sealing application for the type of seal covered by this specification. Sealing surfaces are indicated and the annular space to which the seal assembly must be designed is dimensioned on Figure III. It is further desired that the seals be of double-acting design, that is, capable of withstanding pressure applied from either direction against the seal.

5. DEVELOPMENT PROCEDURE. -

5.1 Phase I. - The seal manufacturer shall develop and deliver to Chance Vought one (1) radial seal assembly of approximately 1 1/2 inch outside diameter and composed of a minimum of four (4) pairs of wedge-type sealing rings designed in accordance with this specification, and also shall submit Phase I report.

5.1.1 The seal assembly shall meet all the requirements of this specification with the exception of the test requirements stated in paragraph 3.

5.1.2 In lieu of the tests specified in paragraph 3, the seal assembly shall meet or exceed the tests specified below, in the order listed, to meet the requirements of Phase I.

(a) proof pressure test at room temperature
(b) 16 hour soak at -65°F
(c) 100 impulse cycles at -65°F
(d) 2000 impulse cycles at +450°F
(e) leakage test at room temperature
(f) 16 hour soak at -65°F
(g) 100 cycles at -65°F
(h) 2000 cycles at +450°F
(i) leakage test at room temperature
(j) proof pressure test per paragraph 3.1

5.1.3 The impulse cycle for the tests listed in paragraph 5.1.2 shall be in accordance with Figure I.

5.2 Phase II. - The seal manufacturer shall investigate, as outlined below, the effects of seal material hardness and varying numbers of seal ring pairs on seal assembly performance and installation torque requirements. In addition, the seal manufacturer shall completely test and qualify the seal assembly described in paragraph 5.2.3 below.

5.2.1 Prior to starting Phase II qualifications testing, the seal manufacturer shall conduct a limited investigation to determine the feasibility of a radial seal

having a hardness not greater than Rockwell C-35, but made to the same nominal dimensions as the seal developed under Phase I. This limited investigation will involve the manufacture of four pairs of wedge seals and static room temperature proof pressure testing to determine torque values required for sealing. The test fixture used for this test shall be the same fixture used for Phase I. The torque values determined herein shall be compared with the torque values determined under similar conditions for the seals having a hardness of Rockwell C-55 to C-60. Chance Vought shall be advised of the results of this investigation prior to continuing the remaining Phase II tests.

5.2.2 The seal manufacturer shall determine the effects on installation torque requirements of increasing the number of seal ring pairs in a radial seal assembly. The test block described in paragraph 5.2.3.2 shall be used for these tests. The seal manufacturer shall determine the torques required to effect proof pressure sealing using 2, 3, 4, 5 and 6 pairs of sealing rings. The sealing rings shall be designed in accordance with paragraph 5.2.3. Spacers of varying lengths shall be used so that the above sequence of torque tests can be run using the same test block.

5.2.2.1 The above torque tests shall be conducted on seal rings made from materials having different hardness properties. One complete set of rings (six pairs) shall be made from material having a hardness not greater than Rockwell C-35. A second set of seals shall be made from material having a hardness of Rockwell C-55 to C-60. The results of these tests shall be forwarded to Chance Vought prior to continuing Phase II testing.

Note: Depending upon the test results of paragraph 5.2.1, the above torque tests may not be required on the seals made from Rockwell C-35 material. Chance Vought shall make this decision when the results of the paragraph 5.2.1 tests are available.

5.2.3 The seal manufacturer shall completely test and qualify a seal assembly of such dimensions as to be suitable for use with the CVP-3352-1 (Class A) Solenoid Selector Valve which is shown in this specification for easy reference. The seal assembly shall be designed with cross-sectional dimensions similar or identical to those developed under Phase I. The seal material shall be the same as that developed under Phase I unless the torque tests of paragraphs 5.2.1 and 5.2.2 indicate that a softer material would be acceptable.

5.2.3.1 Qualification shall consist of all tests specified in paragraph 3 performed on seal assemblies made to minimum and maximum tolerance dimensions. Each seal assembly shall consistof six pairs of sealing rings. The spacers separating the sealing rings shall be designed such that the annular flow area on the outside diameter of the spacer shall be at least equal to that shown by the Class A spacer in the diagram on drawing CVP-3352.

5.2.3.2 The test blocks to be used for qualification shall be designed to resemble as closely as possible the valve installation shown in the diagram on drawing CVP-3352.

5.3 Phase III. - The seal manufacturer shall completely test and qualify a seal assembly of such dimensions as to be suitable for use with the CVP-3352-3 (Class C) Solenoid Selector Valves.

5.3.1 Qualification shall consist of all tests specified in paragraph 3 performed on seal assemblies made to minimum and maximum tolerance dimensions. Each seal assembly shall consist of six pairs of sealing rings identical in design to the rings qualified under Phase II. The spacers separating the sealing rings shall be designed such that the flow restriction into or out of the ports referenced on drawing CVP-3352 shall not be greater than the restriction caused by the Class C spacer shown on CVP-3352.

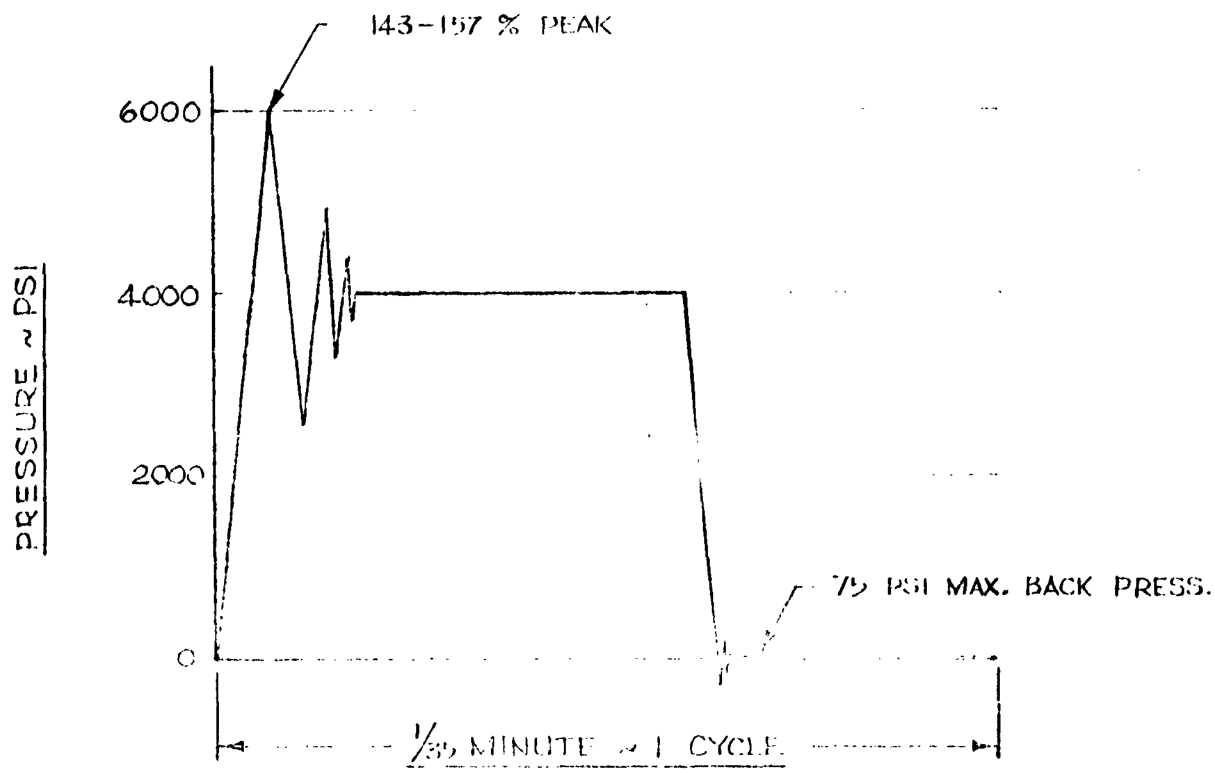
5.3.2 The seal manufacturer shall furnish part numbers, engineering drawings and recommended installation data for sealing rings and spacers of the two nominal seal sizes qualified under this specification. In addition, similar information shall be furnished for seals and spacers of such dimensions as to be suitable for use with the CVP-3352-2 (Class B) Solenoid Selector Valve.

6. ADDITIONAL DATA REQUIRED. -

6.1 Reports. - The seal manufacturer shall submit a final report on all work accomplished under each phase of the development procedure outlined in paragraph 5.

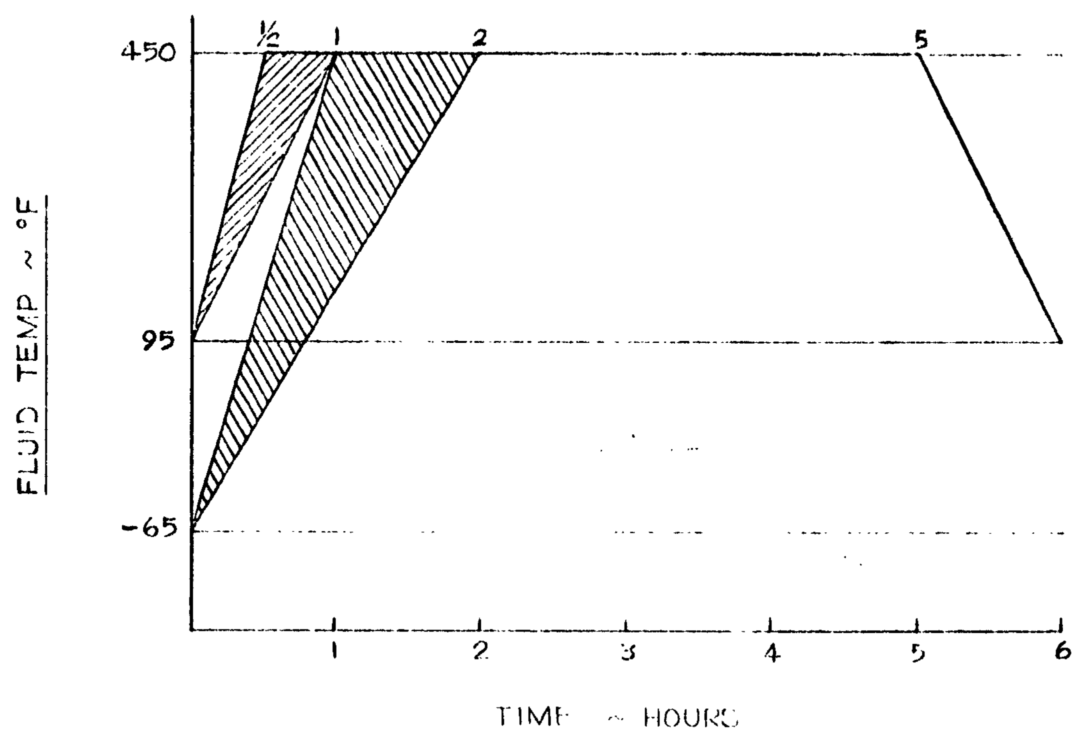
6.1.1 In addition to the final reports specified above, the seal manufacturer shall submit bi-weekly progress reports in letter form.

6.1.2 At any time in the development program, if it becomes apparent that the wedge seal concept will not meet the minimum requirements as outlined in this specification, all work shall be stopped and Chance Vought shall be notified immediately.



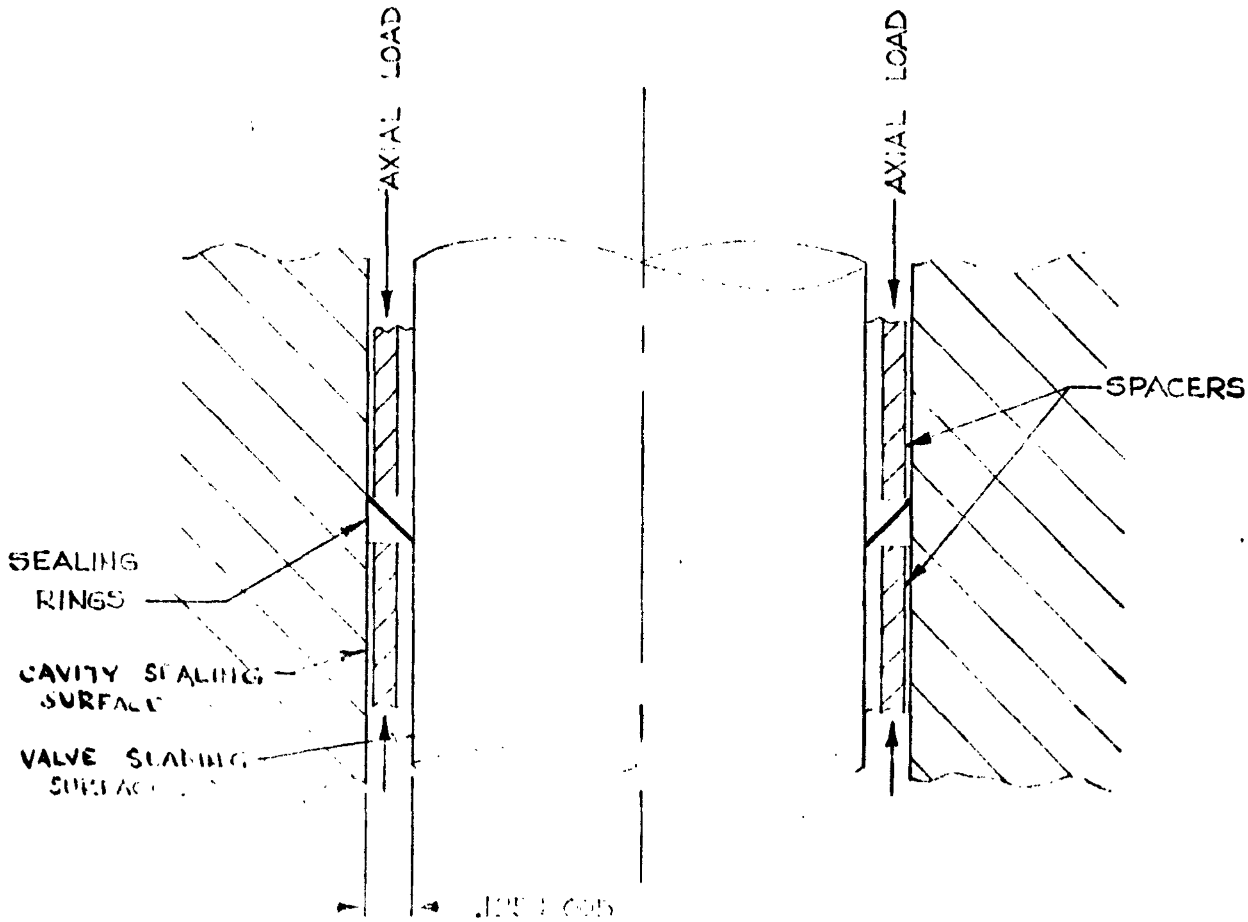
Approximate pressure-time cycle for
impulse testing static metallic seals.

FIGURE I



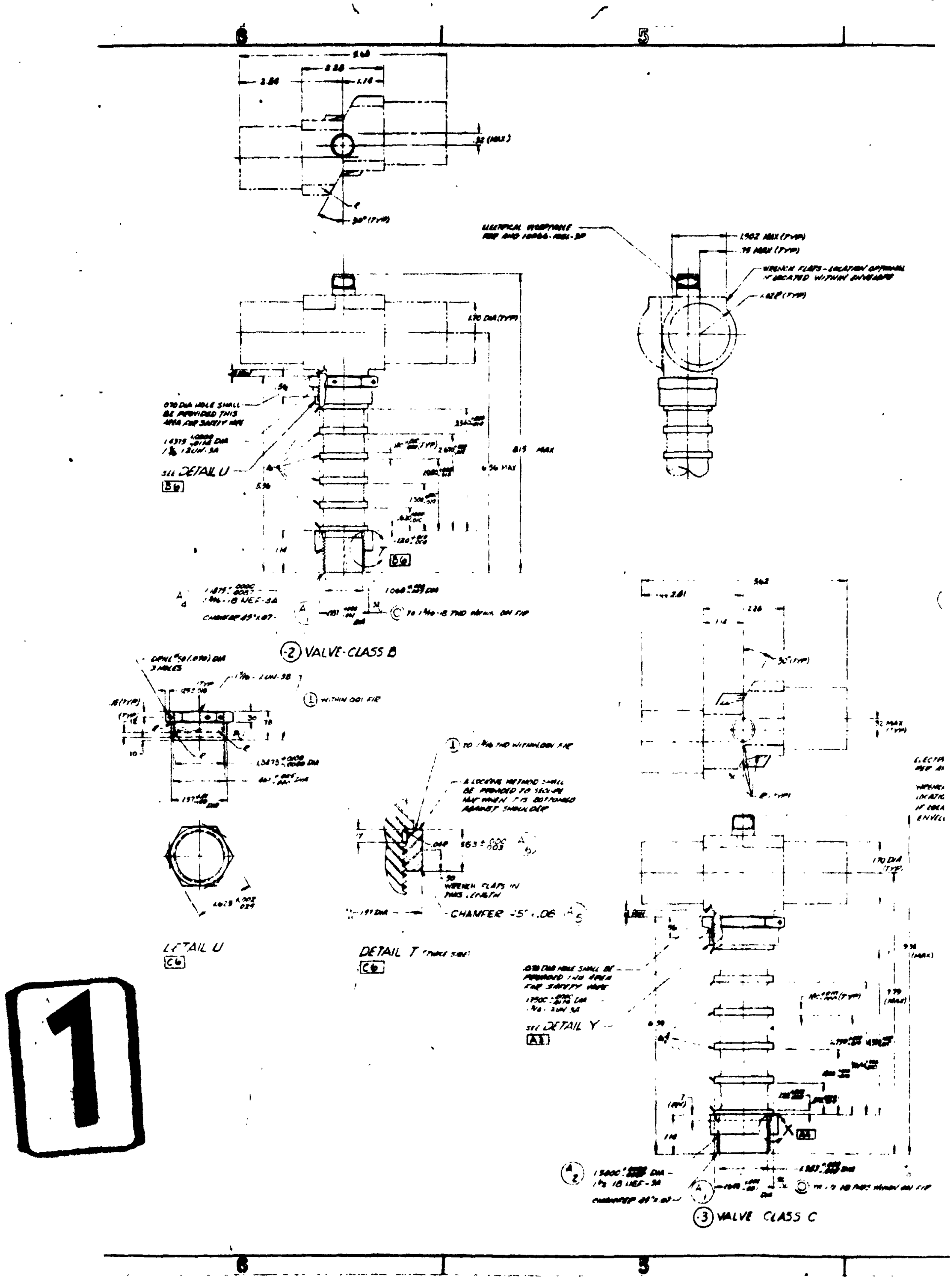
Temperature-time spectrum for
impulse testing static metallic seals.

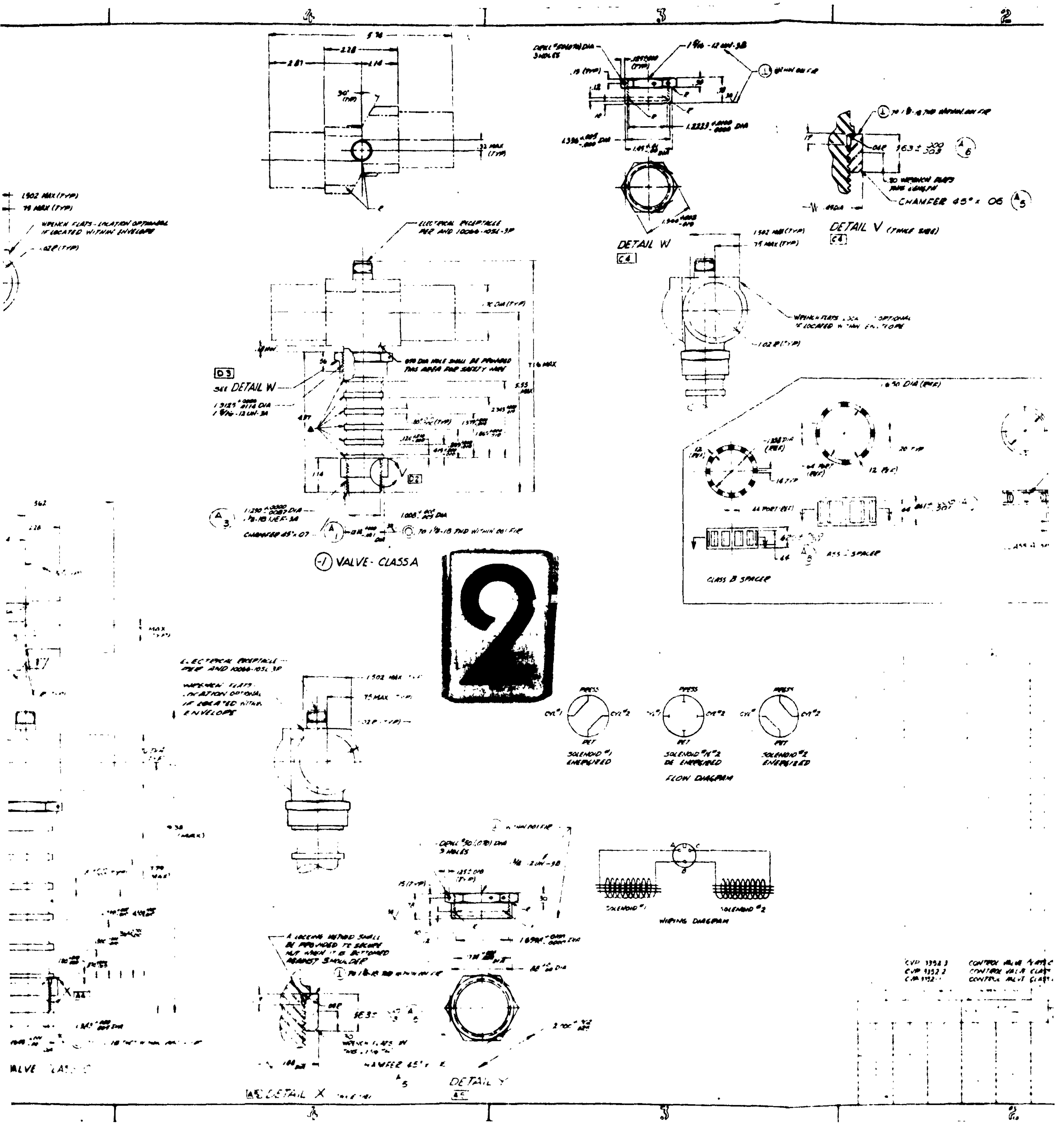
FIGURE II



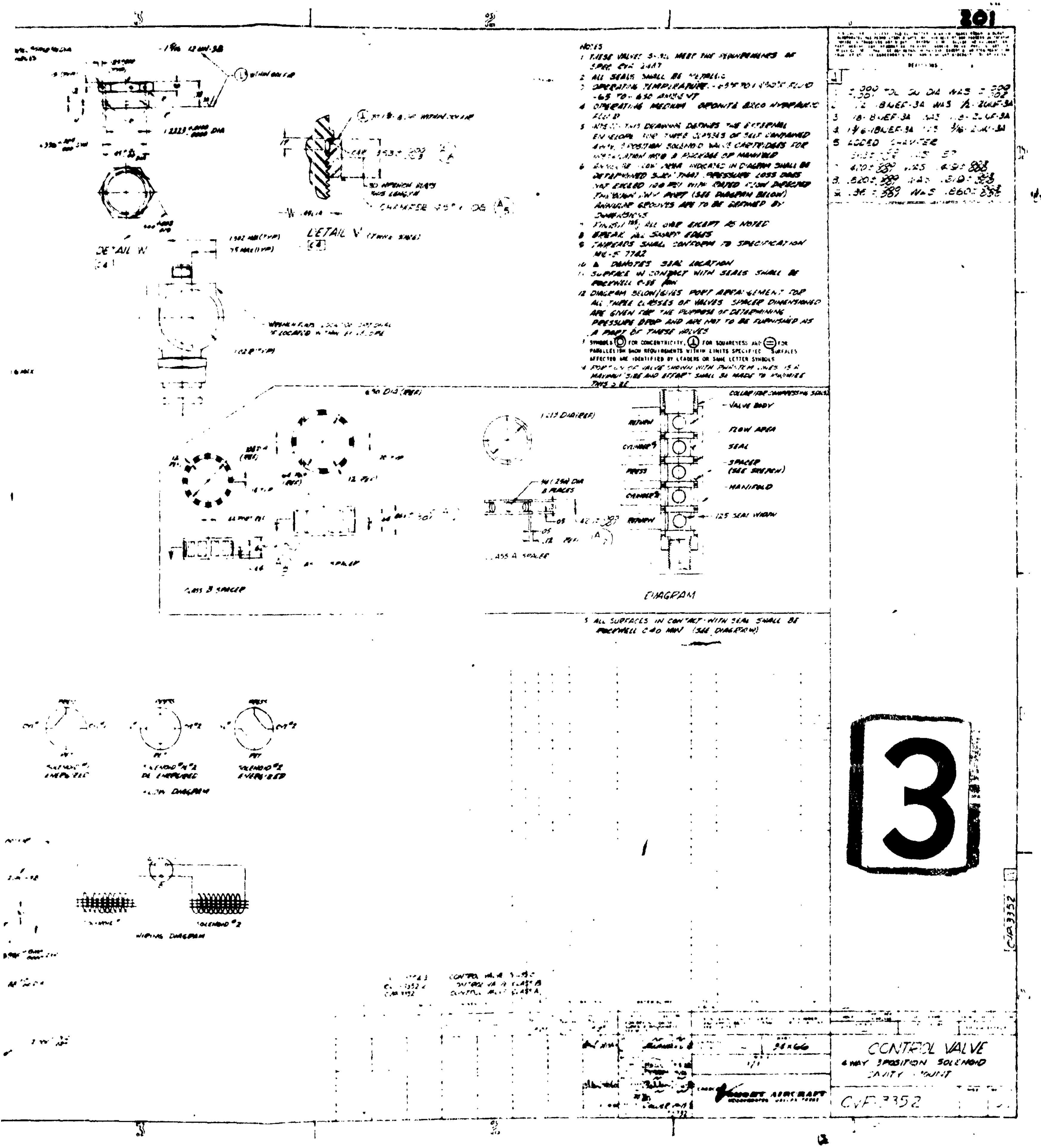
Radial Seal Installation

FIGURE III


6
5
2.10
2.84
1.16
30° (TYP)
ELECTRICAL RECEPTACLE
.070 DIA HOLE SHALL BE PROVIDED THIS AREA FOR SAFETY WIRE
SEE DETAIL U
815 MAX
6.56 MAX
1.0600
.170 DIA (TYP)
1.902 MAX (TYP)
.75 MAX (TYP)
WRENCH FLATS-LOCATION OPTIONAL IF LOCATED WITHIN ENVELOPE
1 3/16-18 UNEF-3A
CHAMFER 45°x.07
2 VALVE-CLASS B
1 WITHIN .001 FIR
1 3/16 - 12UN-3B
DETAIL U
A LOCKING METHOD SHALL BE PROVIDED TO SECURE NUT WHEN IT IS BOTTOMED AGAINST SHOULDER
WRENCH FLATS IN THIS LENGTH
.563
CHAMFER 45°x.06
DETAIL T
.562
2.26
1.14
30° (TYP)
.170 DIA (TYP)
.070 DIA HOLE SHALL BE PROVIDED THIS AREA FOR SAFETY WIRE
1 1/2-12UN-3A
SEE DETAIL Y
9.38 (MAX)
7.79 (MAX)
6.90
1.5000
1 1/2-18 UNEF-3A
CHAMFER 45°x.07
3 VALVE CLASS C
1



DETAIL W
DETAIL V
CHAMFER 45° x .06
SEE DETAIL W
VALVE - CLASS A
CLASS B SPACER
2
PRESS
SOLENOID #1 ENERGIZED
FLOW DIAGRAM
WIRING DIAGRAM
SOLENOID #1
SOLENOID #2
DETAIL X
DETAIL Y


201
NOTES
1 THESE VALVES SHALL MEET THE REQUIREMENTS OF SPEC CVA [illegible]
2 ALL SEALS SHALL BE METALLIC
3 OPERATING TEMPERATURE -65°F TO [illegible] FLUID -65 TO +650 AMBIENT
4 OPERATING MEDIUM OPONITE [illegible] HYDRAULIC FLUID
5 INTENT THIS DRAWING DEFINES THE EXTERNAL ENVELOPE FOR THREE CLASSES OF SELF CONTAINED 4 WAY 3 POSITION SOLENOID VALVE CARTRIDGES FOR INSTALLATION INTO A PACKAGE OR MANIFOLD
6 [illegible] FLOW AREA INDICATED IN DIAGRAM SHALL BE DETERMINED SUCH THAT PRESSURE LOSS DOES NOT EXCEED 100 PSI WITH RATED FLOW [illegible] THE [illegible] PORT (SEE DIAGRAM BELOW) ANNULAR GROOVES ARE TO BE DEFINED BY DIMENSIONS
7 FINISH [illegible] ALL OVER EXCEPT AS NOTED
8 BREAK ALL SHARP EDGES
9 THREADS SHALL CONFORM TO SPECIFICATION MIL-S 7742
10 ▲ DENOTES SEAL LOCATION
11 SURFACE IN CONTACT WITH SEALS SHALL BE ROCKWELL C-38 MIN
12 DIAGRAM BELOW GIVES PORT ARRANGEMENT FOR ALL THREE CLASSES OF VALVES. SPACER DIMENSIONED ARE GIVEN FOR THE PURPOSE OF DETERMINING PRESSURE DROP AND ARE NOT TO BE FURNISHED AS A PART OF THESE VALVES
13 SYMBOLS FOR CONCENTRICITY, FOR SQUARENESS AND FOR PARALLELISM SHOW REQUIREMENTS WITHIN LIMITS SPECIFIED. SURFACES AFFECTED ARE IDENTIFIED BY LEADERS OR SAME LETTER SYMBOLS
14 PORTION OF VALVE SHOWN WITH PHANTOM LINES IS A MAXIMUM SIZE AND EFFORT SHALL BE MADE TO MINIMIZE THIS SIZE
5 ALL SURFACES IN CONTACT WITH SEAL SHALL BE ROCKWELL C40 MIN (SEE DIAGRAM)
DETAIL W
DETAIL V
CHAMFER
COLLAR (FOR COMPRESSING SEALS)
VALVE BODY
RETURN
FLOW AREA
CYLINDER
SEAL
PRESS
SPACER (SEE SKETCH)
MANIFOLD
.125 SEAL WIDTH
DIAGRAM
CLASS A SPACER
CLASS B SPACER
FLOW DIAGRAM
SOLENOID #1 ENERGIZED
SOLENOID #2 ENERGIZED
WIRING DIAGRAM
SOLENOID #2
CONTROL VALVE
4 WAY 3 POSITION SOLENOID
CAVITY MOUNT
CHANCE VOUGHT AIRCRAFT
CVA 3352
3

Originated
By: G. E. Humman

ENGINEERING DEPARTMENT SPECIFICATION

PROCUREMENT SPECIFICATION

FOR

METALLIC SEALS

STATIC APPLICATION

1. SCOPE. - This specification states the design, manufacturing, and testing requirements for metallic seals to be used for static sealing applications in a 4000 psi, 450°F hydraulic system. This specification further outlines the procedure for development of the seal desired.

2. APPLICABLE DOCUMENTS. -

2.1 The following documents of the issue in effect on date of invitations for bids form a part of this specification:

Specifications:
Military:
MIL-H-8446
MIL-STD-10
MIL-H-8891

3. REQUIREMENTS. -

3.1 Qualification. - The metallic seals furnished under this specification shall be a product which has been tested and has passed the qualification tests specified herein.

3.2 Materials. - The materials shall conform to applicable government specifications. Materials which are not covered by applicable specifications may be used provided it can be demonstrated that their use will result in a superior product.

3.2.1 The materials shall possess adequate corrosion-resistant characteristics. The materials shall be of high quality to insure consistence to all its characteristics.

3.2.2 The materials shall be compatible with the fluids and temperature ranges designated in this specification.

3.3 Temperature Range. - The metallic seal shall perform its function in fluid temperatures of -65°F to +450°F and in ambient temperatures of -65°F and +650°F.

3.4 Fluid. - The metallic seal shall perform its function in fluids conforming to specification MIL-H-8446.

3.5 Pressure. - The metallic seal shall function at all operating pressures between zero and 4000 psi. The seal shall function at proof pressures of 6000 psi and shall not fail at burst pressures of 10,000 psi.

3.6 Design and Construction. -

3.6.1 Shape and Dimensions. - The general shape and the general dimensions of the metallic seal procured to this specification shall be controlled by the seal cavity dimensions specified herein in Figures IV, V and VI for 0°, 180° one-way and two-way type seals. Different manufacturer's seals may vary in shape and dimensionally so long as they perform their sealing function within the specified seal cavities.

3.6.1.1 The seal shape and dimensions shall be established for each size cavity in Figures VI, VII and VIII, and the dimensions specified on the seal manufacturer's drawing in accordance with specification MIL-STD-10.

3.6.2 Squeeze. - The initial squeeze which the seal is subjected to in any one cavity may vary .008 inch. This variation is a function of the machining tolerances of the manifold and hydraulic component which form the seal cavity. An additional squeeze variation will be the result of seal tolerances. The seal shall perform satisfactorily at any squeeze between minimum and maximum.

3.6.3 Diametral Back-Up. - The diametral back-up which is provided for the seals varies .001 to .003 inch, depending on the seal size (see Figures IV, V, and VI). This variation is a function of machining tolerances on the manifold or hydraulic component as applicable and the metallic seal.

3.6.4 Sealing Surface Roughness. - The metallic seal shall perform its sealing function on cavity sealing surfaces which vary in surface roughness from zero to 16 RHR. (RHR is defined in MIL-STD-10.)

3.6.5 Sealing Surface Hardness. - The metallic seal shall perform its sealing function on cavity sealing surfaces which have a minimum hardness of Rc 34.

3.6.6 Seal Deflection Force. - The linear compressive force to squeeze the one-way metallic seals (Figures IV and V) shall not exceed 200 pounds per inch of seal mean circumference. For double seals installed to effect two-way sealing (Figure VI), the linear compressive force shall not exceed 400 pounds per inch of seal mean circumference.

3.6.7 Seal Installation. - Figure III shows a typical installation of intended usage. It shall be noted that the seals are subjected to a twisting action as the hydraulic component is torqued down on the seals to give them their proper sealing squeeze.

3.7 Removability. - The seal shall be easy to remove from the cavity after usage. Special tools to remove seal shall not be required.

3.8 Re-Usability. - Re-usability will depend on the springback of the seal after compression and on the variation of squeeze as described in paragraph 3.6.2. The manufacturer shall investigate seal re-usability and describe recommendations on the seal drawing.

3.9 Manufacturing Method. - The metallic seal shall be either formed or machined. Other methods may be used provided they result in a superior product.

3.10 Coatings. - When coatings such as silverplate or teflon are required to make the seal function satisfactorily, they shall be properly noted on the manufacturer's drawing. Coatings shall conform to government specifications. Coatings which are not covered by applicable specifications may be used provided it can be demonstrated that their use will result in a superior product.

3.10.1 Coatings on the metallic seals which contribute unacceptable contamination into Type III hydraulic systems, defined in specifications MIL-H-8891, shall not be used.

3.11 Workmanship. - Workmanship and finish shall be in accordance with the highest grade practice in manufacturing this kind of product. Manufacturing practices shall be such that the physical properties of the finished product shall be consistent from one seal to the next. The product shall be uniform in quality and condition, clean, smooth and free from foreign materials or defects detrimental to the appearance or performance of the parts.

3.12 Interchangeability. - All seals with the same dash number shall be interchangeable.

3.13 Identification of Product. - No coding or special identification is required on the metallic seal.

3.14 Performance. - The metallic seal shall meet all the performance requirements of paragraph 4.2.3.

4. TEST REQUIREMENTS. -

4.1 General. - The tests described below shall be conducted on two seals of each size and type to be tested. One seal shall be installed with minimum deflection and the other shall be installed with maximum deflection. These requirements are shown in Table I.

4.2 Leakage Test. - Each seal, as noted in Table I, shall be subjected to pressure leakage tests at 10, 50, 100, 500 and 1000, 2500 and 4000 psi. Each pressure shall be applied for 2 minutes with no evidence of seal leakage. When two-way seals are being tested, the leakage tests shall be conducted with pressure applied against one side of the seal and then shall be repeated with pressure applied against the opposite side of the seal. This test shall be performed at room temperature and at +450° ± 15°F prior to the proof pressure test of paragraph 4.3. This test shall be performed at -20°F ± 10°F prior to the impulse tests of paragraph 4.5 and at room temperature after the impulse tests. There shall be no evidence of leakage during any of the leakage tests.

4.3 Proof Pressure. - Each seal shall be subjected to proof pressure of 6000 psi at 450°F for a period of 5 minutes. In the case of two-way seals, pressure shall be applied to one side of the seal for 5 minutes and then applied to the other side of the seal for 5 minutes. There shall be no evidence of seal leakage.

4.4 Burst Pressure. - The seals shall withstand a burst pressure of 10,000 psi at room temperature for a period of 5 minutes with no leakage. In the case of two-way seals, the burst pressure shall be applied for 5 minutes against one side of the seal and then applied for 5 minutes against the other side of the seal. There shall be no leakage by the seal in either direction. The burst pressure test shall be the last test performed.

4.5 Impulse Test. - The pressure impulse tests shall be conducted at a rate of 35 ± 5 CPM. Each impulse cycle shall constitute a rise from zero psi to impulse pressure and drop to 4000 psi and then zero psi as shown in Figure I. Hydraulic fluid shall be used as a test medium with a peak impulse pressure of 1.43 to 1.57 times the rated 4000 psi pressure as recorded by an oscilliscope. Each seal shall be tested for a total of 200,000 pressure impulse cycles. The impulse cycles shall be imposed in accordance with paragraph 4.6 below. In the case of two-way seals, pressure impulses shall be applied alternately to each side of the seal until 100,000 impulse cycles have been applied to each side of the seal for a total of 200,000 pressure impulse cycles. After 25,000, 50,000, 100,000, 150,000 and 200,000 pressure impulse cycles, all seals shall be proof pressure tested per paragraph 4.3. There shall be no leakage during the static proof pressure tests. An acceptable method shall be established for measuring leakage during impulse cycling. A leakage not exceeding a rate of 5cc per 1800 impulse cycles is acceptable.

4.6 Temperature-Time Spectrum. - The impulse test (200,000 cycles) shall be conducted while the seal undergoes the temperature-time spectrum shown on Figure II. Each spectrum should take approximately 6 hours to complete and will consist of 11,120 cycles. Eighteen days are required for the test. The first, third, fifth, tenth, thirteenth, and sixteenth spectrums shall begin after the test set-up has soaked at -65°F for 8 hours (overnight). The remaining spectrums shall begin at 95 ± 25°F on the 2nd, 4th, 6th, 7th, 8th, 9th, 11th, 12th, 14th, 15th, 17th, and 18th days of the test. The rate of temperature rise shall be within the shaded areas shown on Figure II.

5. DEVELOPMENT PROCEDURE. -

5.1 General. - The seals described in this specification shall be developed by the seal manufacturer in accordance with the phases outlined in the following paragraphs.

5.2 Phase I. - The seal manufacturer shall develop the optimum seal configuration for the one-way and two-way seals. The two-way seal for this phase shall have an outside diameter of approximately 1 inch and the one-way seal shall

have an outside diameter of the smallest size suitable for use with the 1-inch O.D. two-way seal (see Figure III). The optimum seal configuration should have the minimum practical cross-sectional diameter, minimum installation or tightening force, and maximum allowable squeeze. (Note: The cross-sectional diameter is controlled by Figures IV, V, VI.)

5.2.1 Tests shall be conducted on seal samples made in different wall thicknesses and the optimum wall thickness shall be determined for each of the seal types.

5.2.2 The optimum seal configurations shall meet all the requirements of this specification including the test requirements of paragraph 4.

5.2.3 A force-deflection curve for each of the optimum seal configurations developed in this phase will be obtained and forwarded to Chance Vought as soon as it is available.

5.2.4 The seal manufacturer shall furnish five seals of optimum configuration for 0° and 180° one-way and two-way seals, and shall forward these samples with the final report specified in paragraph 6.1.

5.3 Phase II. - The seal manufacturer shall investigate, as outlined below, the effects of seal material strength and varying seal cavity back-up diameters on seal performance, deflection force requirements, and seal removability. In addition, the seal manufacturer shall completely test and qualify the seals described in paragraph 5.3.2 below. Figure II shows the application of the seals to which they shall be designed.

5.3.1 Prior to starting Phase II qualification testing, the seal manufacturer shall conduct a limited investigation to verify, or re-define, the optimum seal configuration as determined under Phase I. This limited investigation will involve the testing of 0° one-way, 180° one-way, and two-way seals installed with varying back-up diameters. The seals shall meet the deflection force requirements of paragraph   and shall meet all static fluid pressure tests specified in paragraph 4. An additional consideration for optimization is that the seal should be easily removable from a cavity installation after the application of proof pressure when installed at maximum squeeze of .020 inch.

5.3.1.1 The investigation of paragraph 5.3.1 above shall be conducted on seals of each nominal diameter to be qualified under paragraph 5.3.2. Chance Vought shall be notified of the results of this investigation prior to continuing the remaining Phase II tests. Notification shall consist of fluid pressure test results and a force-deflection curve for each optimum seal configuration of each nominal size tested.

5.3.2 The seal manufacturer shall completely test and qualify 0° one-way, 180° one-way, and two-way seals in each of the three nominal inside diameters

listed below. The cavities into which the various types of seals shall be installed for qualification shall be dimensioned in accordance with the seal cavity charts listed in Figures V, VI, VII.

Seal Type	Nom. I.D. for Test	Seal Chart
0° one-way	3/16, 1, 3	Figure V
180° one-way	3/16, 1, 3	Figure VI
Two-way	3/16, 1, 3	Figure VII

5.3.2.1 Qualification shall consist of all tests specified in paragraph 4 performed on seal installations of minimum and maximum tolerance conditions. The minimum tolerance condition is defined as a seal of maximum back-up diameter installed in a seal cavity having a minimum back-up diameter. Conversely, the maximum tolerance condition is defined as a seal of minimum back-up diemeter installed in a cavity having a maximum back-up diameter. Two seal installations shall be tested for each tolerance condition. One installation shall have a seal of minimum height installed in a cavity of maximum height to produce minimum squeeze. In the other installation, a seal of maximum height shall be installed in a cavity of minimum height toproduce maximum squeeze.

5.3.2.2 For each nominal diameter of each type of seal being qualified, an identical seal shall be subjected to a force-deflection test. These seals shall meet the deflection force requirements of paragraph 4.6.6. A force-deflection curve for each type and nominal diameter seal which is qualified shall be submitted to Chance Vought along with the Phase II report.

5.3.2.3 The table below summarizes the testing which is required under paragraph 5.3.2 for each of the three seal types covered by Phase II. A total of 45 seals shall be tested. The test block seal seat material shall be 17-4PH, 416, or equivalent.

TABLE I

SEALS TO BE TESTED			NOMINAL I.D.	TOLERANCE CONDITION	SQUEEZE CONDITION	FORCE DEFLECTION TEST	HYDRAULIC TESTS
0°	180°	2-Way					
1	16	31	3/16	Nominal	--	X	
2	17	32		Maximum	Maximum		X
3	18	33			Minimum		X
4	19	34		Minimum	Maximum		X
5	20	35			Minimum		X
6	21	36	1	Nominal	--	X	
7	22	37			Maximum		X
8	23	38		Maximum	Minimum		X
9	24	39			Maximum		X
10	25	40		Minimum	Minimum		X

TABLE I (Continued)

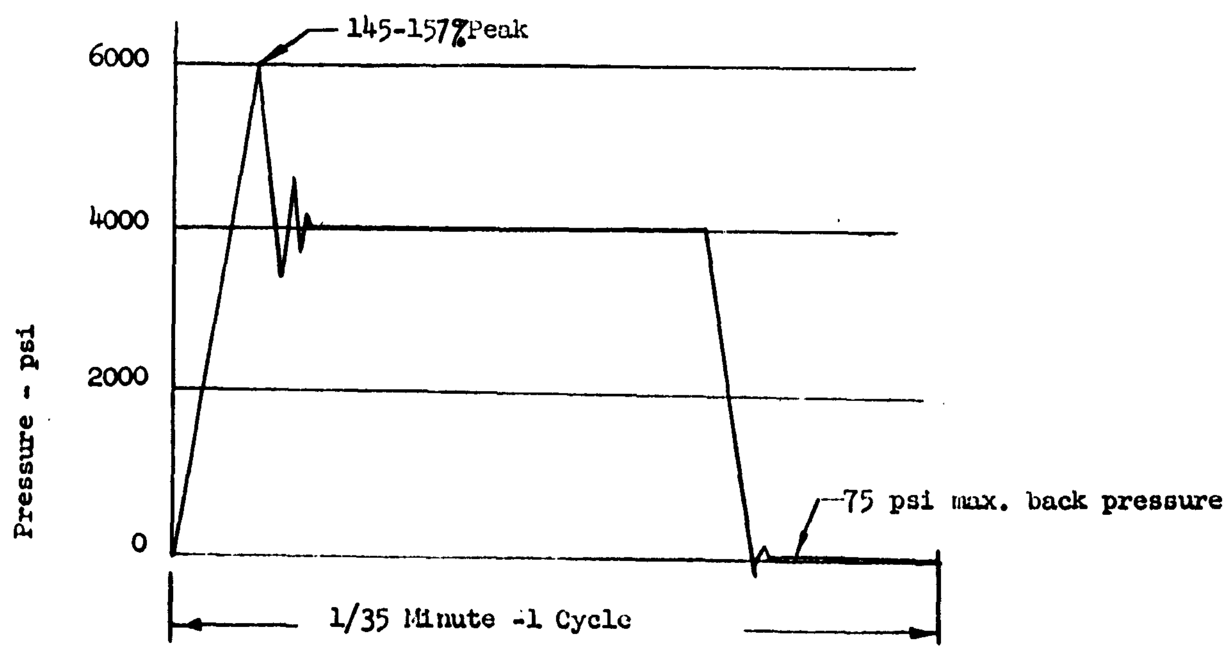
SEALS TO BE TESTED			NOMINAL I.D.	TOLERANCE CONDITION	SQUEEZE CONDITION	FORCE DEFLECTION TEST	HYDRAULIC TESTS
0°	180°	2-Way					
11	26	41	3	Nominal	--	X	
12	27	42		Maximum	Maximum		X
13	28	43			Minimum		X
14	29	44		Minimum	Maximum		X
15	30	45			Minimum		X

6. ADDITIONAL DATA REQUIRED. -

6.1 Reports. - Included in the work for each phase is a final report to be submitted by the seal manufacturer on all work accomplished under each phase of the development procedure outlined in paragraph 5.

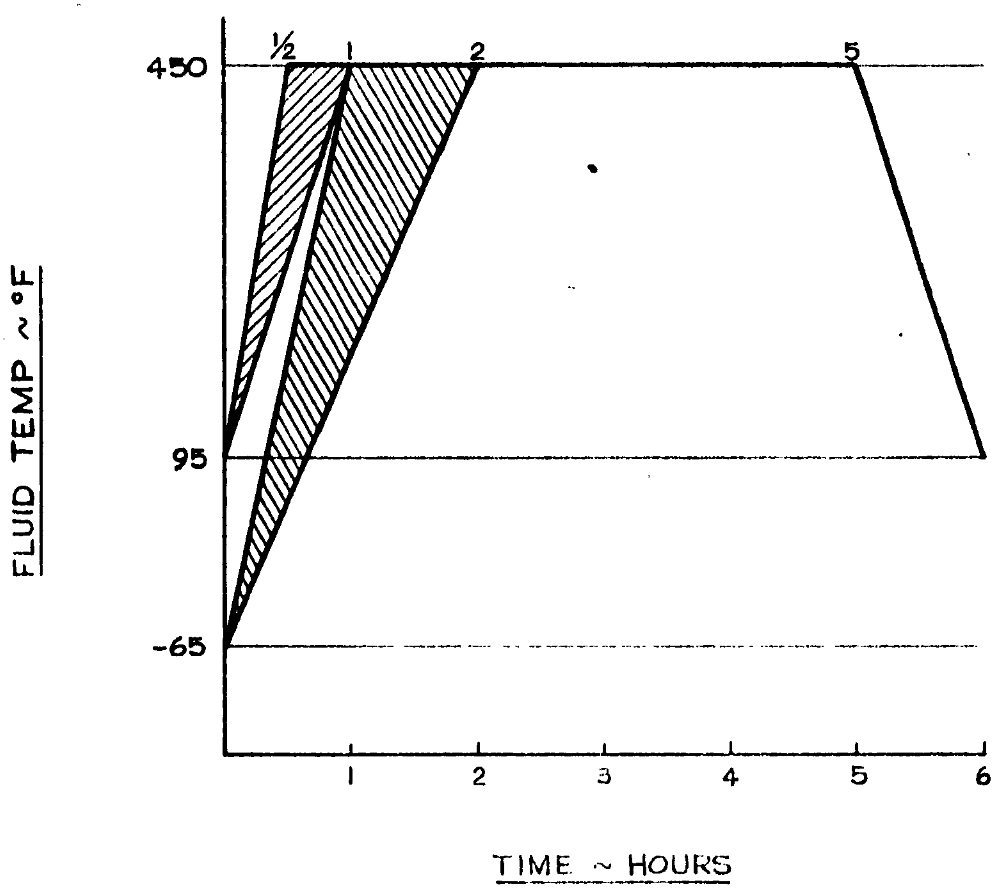
6.1.1 In addition to the final reports specified above, the seal manufacturer shall submit bi-weekly progress reports in letter form.

6.1.2 At any time in the development program, if it becomes apparent that the seal concept will not meet the minimum requirements of this specification, all work shall be stopped and Chance Vought shall be notified immediately.



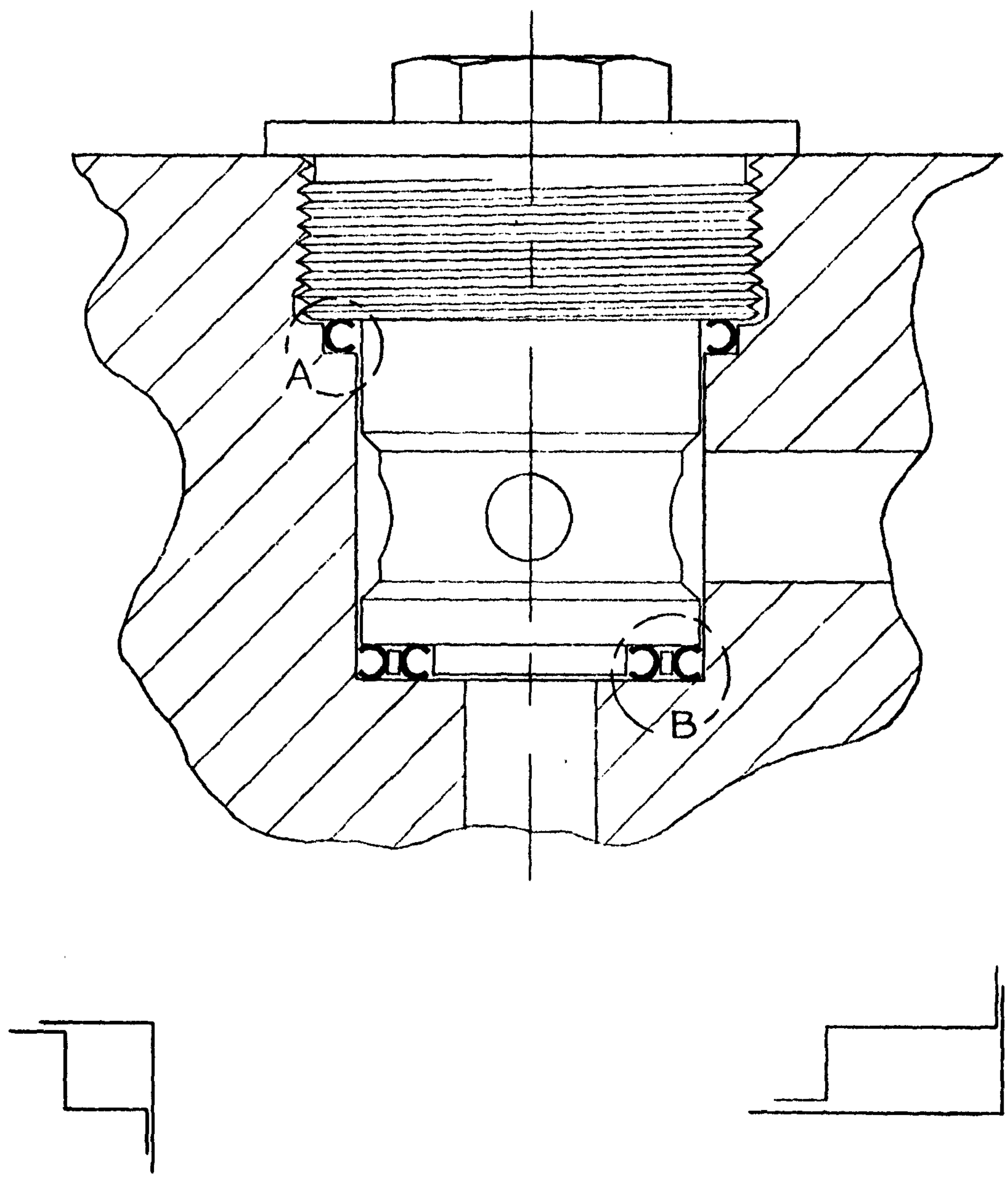
Approximate pressure time cycle for impulse testing static metallic seals

FIGURE I



Temperature-time spectrum for impulse testing static metallic seals.

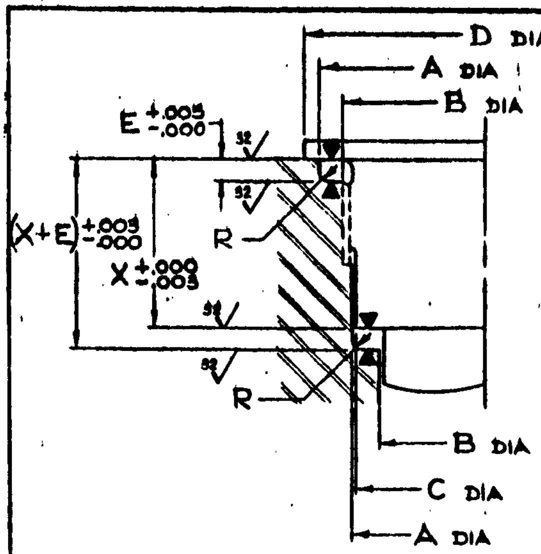
FIGURE 11



DETAIL A
ONE - WAY SEAL CAVITY

DETAIL B
TWO - WAY SEAL CAVITY

FIGURE III
SEAL USAGE

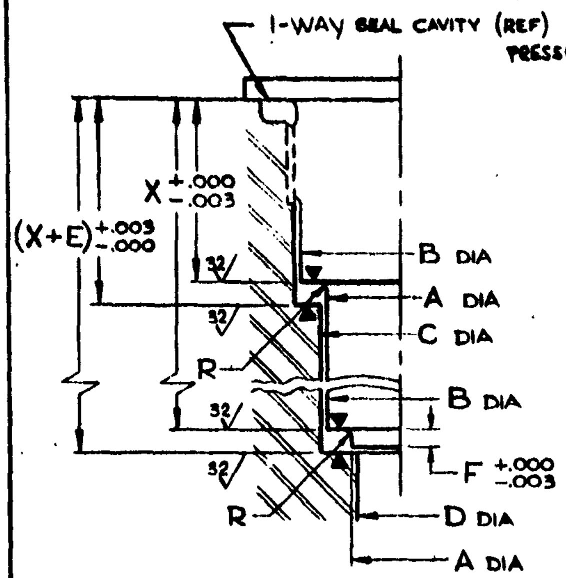


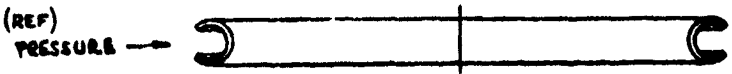
0° ONE-WAY SEAL

1. THIS CHART TO BE USED FOR ONE-WAY 0° SEALS ONLY.
2. SYMBOL ▼ DENOTES SEALING SURFACES.
3. "X" REPRESENTS THE NOMINAL LINEAR DIMENSION TO A VALVE SEALING SURFACE.
4. "X+E" REPRESENTS THE NOMINAL DEPTH DIMENSION TO A VALVE CAVITY SEALING SURFACE.
5. THE VALVE SEALING SURFACES SHALL BE PARALLEL WITHIN .002 F.I.R. AND PERPENDICULAR TO THE THREAD AXIS WITHIN .003 F.I.R.
6. THE VALVE CAVITY SEALING SURFACES SHALL BE PARALLEL WITHIN .002 F.I.R. AND PERPENDICULAR TO THE THREAD AXIS WITHIN .003 F.I.R.
7. ALL SEALING SURFACES SHALL HAVE A MINIMUM HARDNESS OF ROCKWELL C-35.

DASH NO.	NOM I.D.	A DIA		B DIA MAX	C DIA		D DIA MIN	E	R MAX
-0003	3/16	.375		.190	.370		.52	.079	.03
-0732	7/32	.406		.221	.401		.55		
-0004	1/4	.437		.252	.432		.58		
-0932	9/32	.468		.283	.463		.61		
-0005	5/16	.500		.315	.495		.65		
-1132	11/32	.531		.346	.526		.68		
-0006	3/8	.562		.377	.557		.71		
-1332	13/32	.593		.408	.588		.74		
-0007	7/16	.625	+.001 -.000	.440	.620	+.000 -.002	.77		
-1532	15/32	.656		.471	.651		.80		
-0008	1/2	.687		.502	.682		.83		
-1732	17/32	.718		.533	.713		.86		
-0009	9/16	.750		.565	.745		.90		
-1932	19/32	.781		.596	.776		.93		
-0010	5/8	.812		.627	.807		.96		
-2132	21/32	.843		.658	.838		.99		
-0011	11/16	.875		.690	.870		1.02		
-2332	23/32	.906		.721	.901		1.05		
-0012	3/4	.937		.752	.932		1.08		
-2532	25/32	.968		.783	.963		1.11		
-0013	13/16	1.000		.815	.995		1.15		
-2732	27/32	1.031		.846	1.026		1.18		
-0014	7/8	1.062		.877	1.057		1.21		
-2932	29/32	1.093		.908	1.088		1.24		
-0015	15/16	1.125		.940	1.120		1.27		
-3132	31/32	1.156		.971	1.151		1.30		
-0016	1	1.187		1.002	1.182		1.33		
-0017	1 1/16	1.250		1.065	1.245		1.40		
-0018	1 1/8	1.312	+.002 -.000	1.127	1.307		1.46		
-0019	1 3/16	1.375		1.190	1.370		1.52		
-0020	1 1/4	1.437		1.252	1.432		1.58		
-0021	1 5/16	1.500		1.315	1.495		1.65		
-0022	1 3/8	1.562		1.377	1.557		1.71		
-0023	1 7/16	1.625		1.440	1.620		1.77		
-0024	1 1/2	1.687		1.502	1.682		1.83		
-0025	1 9/16	1.750		1.565	1.745		1.90		
-0026	1 5/8	1.812		1.627	1.807		1.96		
-0027	1 11/16	1.875		1.690	1.870		2.02		
-0028	1 3/4	1.937		1.752	1.932		2.08		
-0029	1 13/16	2.000		1.815	1.995		2.15		
-0030	1 7/8	2.062		1.877	2.057		2.21		
-0031	1 15/16	2.125		1.940	2.120		2.27	.079	.03
-0032	2	2.250		2.015	2.245		2.40	.111	.05
-0033	2 1/16	2.312		2.077	2.307		2.46		
-0034	2 1/8	2.375		2.140	2.370		2.52		
-0035	2 3/16	2.437		2.202	2.432		2.58		
-0036	2 1/4	2.500		2.265	2.495		2.65		
-0037	2 5/16	2.562		2.327	2.557		2.71		
-0038	2 3/8	2.625		2.390	2.620		2.77		
-0039	2 7/16	2.687	+.003 -.000	2.452	2.682	+.000 -.002	2.83		
-0040	2 1/2	2.750		2.515	2.745		2.90		
-0041	2 9/16	2.812		2.577	2.807		2.96		
-0042	2 5/8	2.875		2.640	2.870		3.02		
-0043	2 11/16	2.937		2.702	2.932		3.08		
-0044	2 3/4	3.000		2.765	2.995		3.15		
-0045	2 13/16	3.062		2.827	3.057		3.21		
-0046	2 7/8	3.125		2.890	3.120		3.27		
-0047	2 15/16	3.187		2.952	3.182		3.33		
-0048	3	3.250		3.015	3.245		3.40	.111	.05

Figure IV



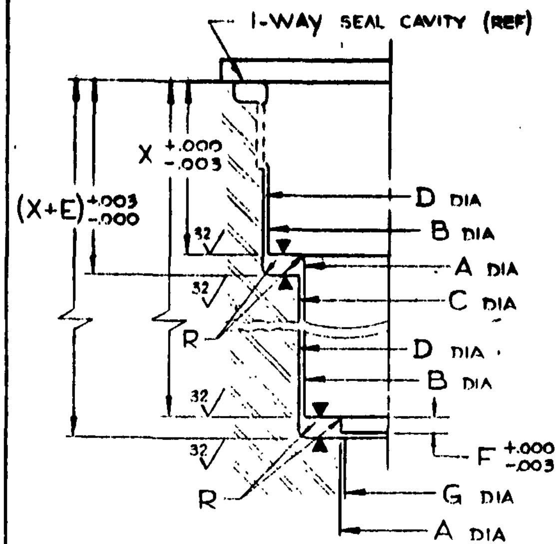


180° ONE-WAY SEAL

1. THIS CHART TO BE USED FOR ONE WAY 180° SEALS ONLY.
2. SYMBOL ▼ DENOTES SEALING SURFACES.
3. "X" REPRESENTS THE NOMINAL LINEAR DIMENSION TO A VALVE SEALING SURFACE.
4. "X+E" REPRESENTS THE NOMINAL DEPTH DIMENSION TO A VALVE CAVITY SEALING SURFACE.
5. THE VALVE SEALING SURFACES SHALL BE PARALLEL WITHIN .002 F.I.R. AND PERPENDICULAR TO THE THREAD AXIS WITHIN .003 F.I.R.
6. THE VALVE CAVITY SEALING SURFACES SHALL BE PARALLEL WITHIN .002 F.I.R. AND PERPENDICULAR TO THE THREAD AXIS WITHIN .003 F.I.R.
7. ALL SEALING SURFACES SHALL HAVE A MINIMUM HARDNESS OF ROCKWELL C-35.

DASH NO.	NOM I.D.	A DIA		B DIA MIN	C DIA		D DIA MAX	E	F	R MAX
-0003	3/16	.188		.373	.193		.178	.079	.074	.03
-0732	7/32	.219		.404	.224		.209			
-0004	1/4	.250		.435	.255		.240			
-0932	9/32	.281		.466	.286		.271			
-0005	5/16	.313		.498	.318		.303			
-1132	11/32	.344		.529	.349		.334			
-0006	3/8	.375		.560	.380		.365			
-1332	13/32	.406		.591	.411		.396			
-0007	7/16	.438		.623	.443		.428			
-1532	15/32	.469		.654	.474		.459			
-0008	1/2	.500		.685	.505		.490			
-1732	17/32	.531		.716	.536		.521			
-0009	9/16	.563		.748	.568		.553			
-1932	19/32	.594	+.000 -.001	.779	.599		.584			
-0010	5/8	.625		.810	.630		.615			
-2132	21/32	.656		.841	.661		.646			
-0011	11/16	.688		.873	.693		.678			
-2332	23/32	.719		.904	.724		.709			
-0012	3/4	.750		.935	.755		.740			
-2532	25/32	.781		.966	.786		.771			
-0013	13/16	.813		.998	.818	+.002 -.000	.803			
-2732	27/32	.844		1.029	.849		.834			
-0014	7/8	.875		1.060	.880		.865			
-2932	29/32	.906		1.091	.911		.896			
-0015	15/16	.938		1.123	.943		.928			
-3132	31/32	.969		1.154	.974		.959			
-0016	1	1.000		1.185	1.005		.990			
-0017	1 1/16	1.063		1.248	1.068		1.053			
-0018	1 1/8	1.125		1.310	1.130		1.115			
-0019	1 3/16	1.188		1.373	1.193		1.178			
-0020	1 1/4	1.250		1.435	1.255		1.240			
-0021	1 5/16	1.313		1.498	1.318		1.303			
-0022	1 3/8	1.375		1.560	1.380		1.365			
-0023	1 7/16	1.438		1.623	1.443		1.428			
-0024	1 1/2	1.500		1.685	1.505		1.490			
-0025	1 9/16	1.563		1.748	1.568		1.553			
-0026	1 5/8	1.625		1.810	1.630		1.615			
-0027	1 11/16	1.688		1.873	1.693		1.678			
-0028	1 3/4	1.750		1.935	1.755		1.740			
-0029	1 13/16	1.813		1.998	1.818		1.803			
-0030	1 7/8	1.875		2.060	1.880		1.865			
-0031	1 15/16	1.938		2.123	1.943		1.928	.079	.074	.03
-0032	2	2.000	+.000 -.002	2.237	2.005		1.990	.111	.106	.05
-0033	2 1/16	2.063		2.300	2.068		2.053			
-0034	2 1/8	2.125		2.362	2.130		2.115			
-0035	2 3/16	2.188		2.425	2.193		2.178			
-0036	2 1/4	2.250		2.487	2.255		2.240			
-0037	2 5/16	2.313		2.550	2.318		2.303			
-0038	2 3/8	2.375		2.612	2.380		2.365			
-0039	2 7/16	2.438		2.675	2.443		2.428			
-0040	2 1/2	2.500		2.737	2.505	+.003 -.000	2.490			
-0041	2 9/16	2.563		2.800	2.568		2.553			
-0042	2 5/8	2.625		2.862	2.630		2.615			
-0043	2 11/16	2.688		2.926	2.693		2.676			
-0044	2 3/4	2.750		2.987	2.755		2.740			
-0045	2 13/16	2.813		3.050	2.818		2.803			
-0046	2 7/8	2.875		3.112	2.880		2.865			
-0047	2 15/16	2.938		3.175	2.943		2.928			
-0048	3	3.000		3.237	3.005		2.990	.111	.106	.05

Figure V



TWO-WAY SEAL

1. THIS CHART TO BE USED FOR TWO-WAY SEALS ONLY.
2. SYMBOL ▼ DENOTES SEALING SURFACES.
3. "X" REPRESENTS THE NOMINAL LINEAR DIMENSION TO A VALVE SEALING SURFACE.
4. "X+E" REPRESENTS THE NOMINAL DEPTH DIMENSION TO A VALVE CAVITY SEALING SURFACE.
5. THE VALVE SEALING SURFACES SHALL BE PARALLEL WITHIN .002 F.I.R. AND PERPENDICULAR TO THE THREAD AXIS WITHIN .003 F.I.R.
6. THE VALVE CAVITY SEALING SURFACES SHALL BE PARALLEL WITHIN .002 F.I.R. AND PERPENDICULAR TO THE THREAD AXIS WITHIN .003 F.I.R.
7. ALL SEALING SURFACES SHALL HAVE A MINIMUM HARDNESS OF ROCKWELL C-35.

DASH NO.	NOM I.D.	A DIA		B DIA		C DIA		D DIA		E	F	G DIA MAX	R MAX
-0003	3/16	.188		.557		.193		.562		.079	.074	.170	.03
-0732	7/32	.219		.588		.224		.593				.209	
-0004	1/4	.250		.620		.255		.625				.240	
-0932	9/32	.281		.651		.286		.656				.271	
-0005	5/16	.313		.682		.318		.687				.303	
-1132	11/32	.344		.713		.349		.718				.334	
-0006	3/8	.375		.745		.380		.750				.365	
-1332	13/32	.406		.776		.411		.781				.396	
-0007	7/16	.438		.807		.443		.812				.428	
-1532	15/32	.469		.838		.474		.843				.459	
-0008	1/2	.500		.870		.505		.875				.490	
-1732	17/32	.531		.901		.536		.906				.521	
-0009	9/16	.563		.932		.568		.937				.553	
-1932	19/32	.594	+.000 -.001	.963	+.000 -.002	.599		.968				.584	
-0010	5/8	.625		.995		.630		1.000				.615	
-2132	21/32	.656		1.026		.661		1.031				.646	
-0011	11/16	.688		1.057		.693		1.062				.678	
-2332	23/32	.719		1.088		.724		1.093				.709	
-0012	3/4	.750		1.120		.755		1.125				.740	
-2532	25/32	.781		1.151		.786		1.156				.771	
-0013	13/16	.813		1.182		.818	+.002 -.000	1.187	+.002 -.000			.803	
-2732	27/32	.844		1.213		.849		1.218				.834	
-0014	7/8	.875		1.245		.880		1.250				.865	
-2932	29/32	.906		1.276		.911		1.281				.896	
-0015	15/16	.938		1.307		.943		1.312				.928	
-3132	31/32	.969		1.338		.974		1.343				.959	
-0016	1	1.000		1.370		1.005		1.375				.990	
-0017	1 1/16	1.063		1.432		1.068		1.437				1.053	
-0018	1 1/8	1.125		1.495		1.130		1.500				1.115	
-0019	1 3/16	1.188		1.557		1.193		1.562				1.178	
-0020	1 1/4	1.250		1.620		1.255		1.625				1.240	
-0021	1 5/16	1.313		1.682		1.318		1.687				1.303	
-0022	1 3/8	1.375		1.745		1.380		1.750				1.365	
-0023	1 7/16	1.438		1.807		1.443		1.812				1.428	
-0024	1 1/2	1.500		1.870		1.505		1.875				1.490	
-0025	1 9/16	1.563		1.932		1.568		1.937				1.553	
-0026	1 5/8	1.625		1.995		1.630		2.000				1.615	
-0027	1 11/16	1.688		2.057		1.693		2.062				1.678	
-0028	1 3/4	1.750		2.120		1.755		2.125				1.740	
-0029	1 13/16	1.813		2.182		1.818		2.187				1.803	
-0030	1 7/8	1.875		2.245		1.880		2.250				1.865	
-0031	1 15/16	1.938		2.307		1.943		2.312		.079	.074	1.928	.03
-0032	2	2.000	+.000 -.002	2.495	+.000 -.002	2.005		2.500		.111	.106	1.990	.05
-0033	2 1/16	2.063		2.557		2.068		2.562				2.053	
-0034	2 1/8	2.125		2.620		2.130		2.625				2.115	
-0035	2 3/16	2.188		2.682		2.193		2.687				2.178	
-0036	2 1/4	2.250		2.745		2.255		2.750				2.240	
-0037	2 5/16	2.313		2.807		2.318		2.812				2.303	
-0038	2 3/8	2.375		2.870		2.380		2.875				2.365	
-0039	2 7/16	2.438		2.932		2.443		2.937				2.428	
-0040	2 1/2	2.500		2.995		2.505	+.003 -.000	3.000	+.003 -.000			2.490	
-0041	2 9/16	2.563		3.057		2.568		3.062				2.553	
-0042	2 5/8	2.625		3.120		2.630		3.125				2.615	
-0043	2 11/16	2.688		3.182		2.693		3.187				2.678	
-0044	2 3/4	2.750		3.245		2.755		3.250				2.740	
-0045	2 13/16	2.813		3.307		2.818		3.312				2.803	
-0046	2 7/8	2.875		3.370		2.880		3.375				2.865	
-0047	2 15/16	2.938		3.432		2.943		3.437				2.928	
-0048	3	3.000		3.495		3.005		3.500		.111	.106	2.990	.05

Figure VI

APPENDIX II-6

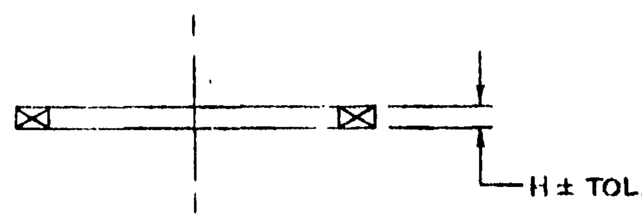
EFFECT OF CAVITY, COMPONENT AND SEAL TOLERANCES ON SQUEEZE

REQUIRED SEAL DEFLECTIONS

THE TABLE BELOW SHOWS THE SEAL DEFLECTIONS REQUIRED FOR SEALS MADE TO VARIOUS TOLERANCES. THE ALLOWABLE SEAL DEFLECTION MUST BE GREAT ENOUGH TO ACCOMMODATE THE TOTAL TOLERANCE SPREAD IN THE SEAL CAVITY AND ON THE SEAL ITSELF PLUS THE MINIMUM DEFLECTION REQUIRED TO INSURE PROPER SEALING.

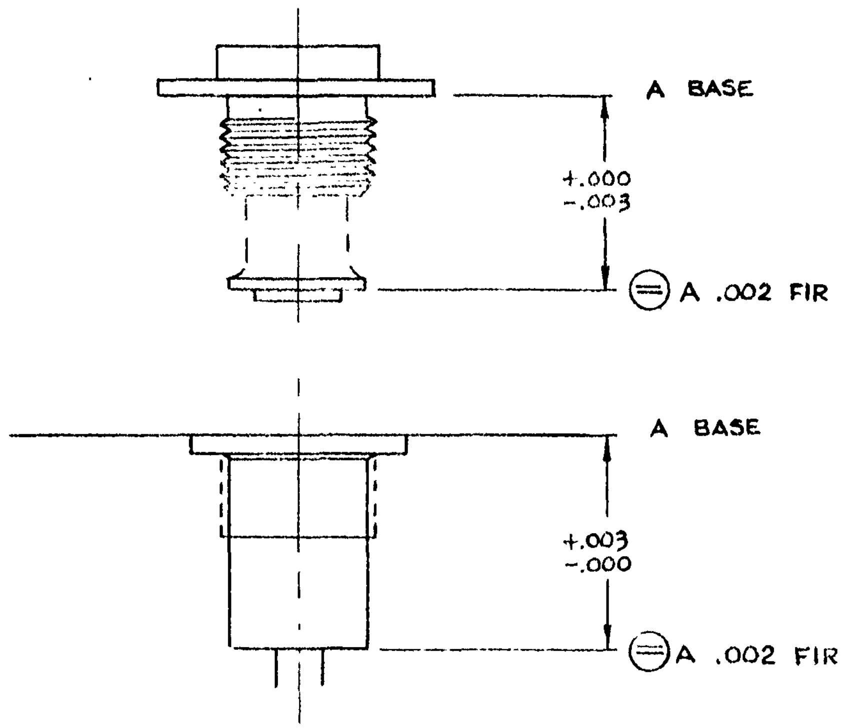
ASSUME MINIMUM DEFLECTION REQD = .004

TOTAL DEFLECTION = TOTAL TOLERANCE SPREAD
+ MIN. REQD. DEFLECTION



SEAL TOLERANCE	REQD DEFLECTION
±.001	≥ .016
±.002	.018
±.003	.020
±.004	.022
±.005	.024
±.006	.026
±.007	.028
±.008	.030

SEAL CAVITY TOLERANCE STUDY



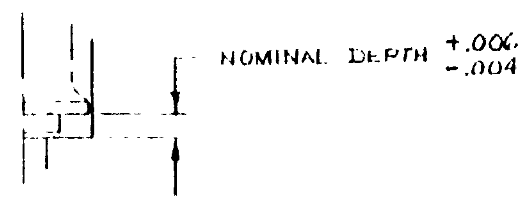
TOTAL TOLERANCES :

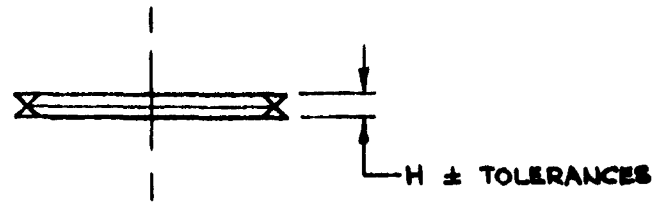
1. MODULE LENGTH -.003
   CAVITY DEPTH +.003

+.006

2. PARALLELISM :
   MODULE -.002
   CAVITY -.002

-.004





ASSUME SEAL TOLERANCES = $\pm .002$

MAX. SEAL HEIGHT = $H + .002 = H'$

MIN. SEAL HEIGHT = $H - .002 = H''$

SEAL CAVITY TOLERANCES = $^{+.006}_{-.004}$ (FROM PRECEDING PAGE)

MAX. CAVITY DEPTH = $D + .006 = D'$

MIN. CAVITY DEPTH = $D - .004 = D''$

MAXIMUM CAVITY DEPTH MUST BE EQUAL TO OR LESS THAN MINIMUM SEAL HEIGHT MINUS MINIMUM SQUEEZE.

$$D' \leqq H'' - \text{MIN. SQUEEZE}$$

MINIMUM CAVITY DEPTH MUST BE EQUAL TO OR GREATER THAN MAXIMUM SEAL HEIGHT MINUS MAXIMUM SQUEEZE.

$$D'' \geqq H' - \text{MAX. SQUEEZE}$$

ASSUME MINIMUM REQUIRED SQUEEZE = .004

TOTAL SQUEEZE REQUIRED = TOTAL TOLERANCE SPREAD + MIN. SQUEEZE

MIN. CAVITY WITH MAX. SEAL -	.006 (CAVITY)	
	.002 (SEAL)	
	.008	.008
MAX. CAVITY WITH MIN. SEAL -	.004 (CAVITY)	
	.002 (SEAL)	
	.006	.006
	TOTAL TOLERANCE SPREAD =	.014
	+ MIN. SQUEEZE +	.004
	TOTAL SQUEEZE REQUIRED =	**.018**

OR :

TOLERANCE VARIATION D′ TO D″ = .010
TOLERANCE VARIATION H′ TO H″ = .004

.014

\+ MIN. SQUEEZE + .004

TOTAL SQUEEZE REQUIRED = .018

ASSUME NOMINAL SEAL HEIGHT = .100 ± .002

CAVITY DEPTH = $.088^{+.006}_{-.004}$

MIN. SQUEEZE = .004

MAX. SQUEEZE = .018

$D' \leqq H'' - .004$

$.094 \leqq .098 - .004$ ✓

$D'' \geqq H' - .018$

$.084 \geqq .102 - .018$ ✓

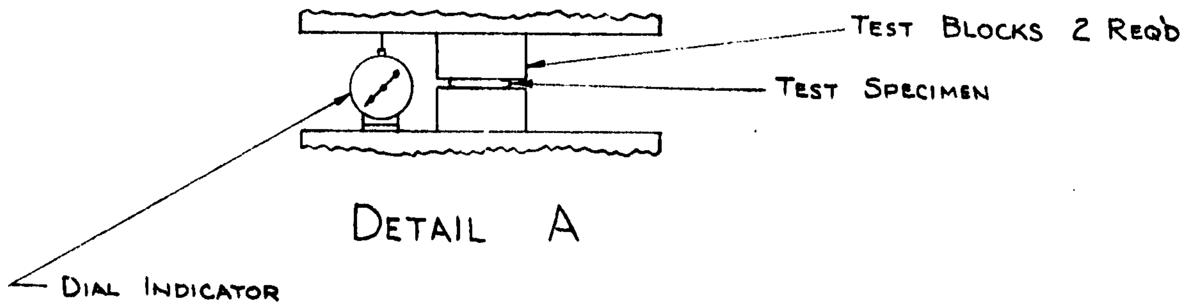
SEAL TOL.	TOTAL TOL. SPREAD +	MIN. SQUEEZE =	REQUIRED SQUEEZE
±.001	.002 + .010	.004	.016
±.002	.004 + .010	.004	.018
±.003	.006 + .010	.004	.020
±.004	.008 + .010	.004	.022
±.005	.010 + .010	.004	.024
±.006	.012 + .010	.004	.026
±.007	.014 + .010	.004	.028
±.008	.016 + .010	.004	.030

APPENDIX II-7

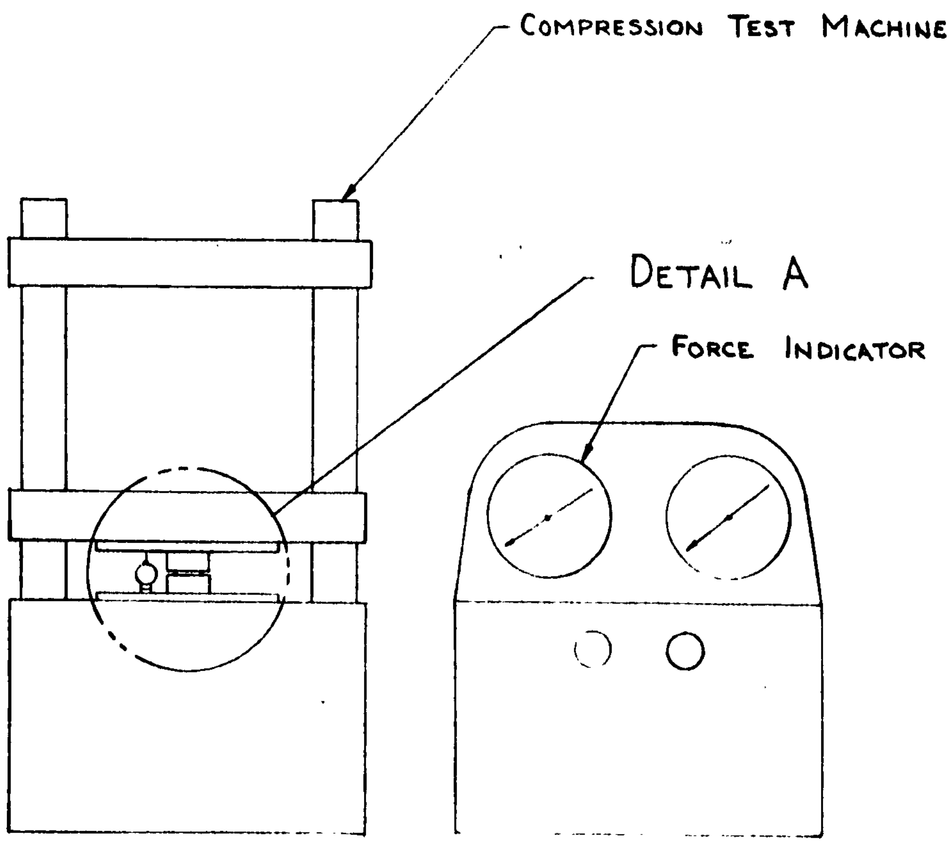
SEAL COMPRESSION TEST SET UP

FORCE VS. DEFLECTION CURVES FOR HI-CEAL

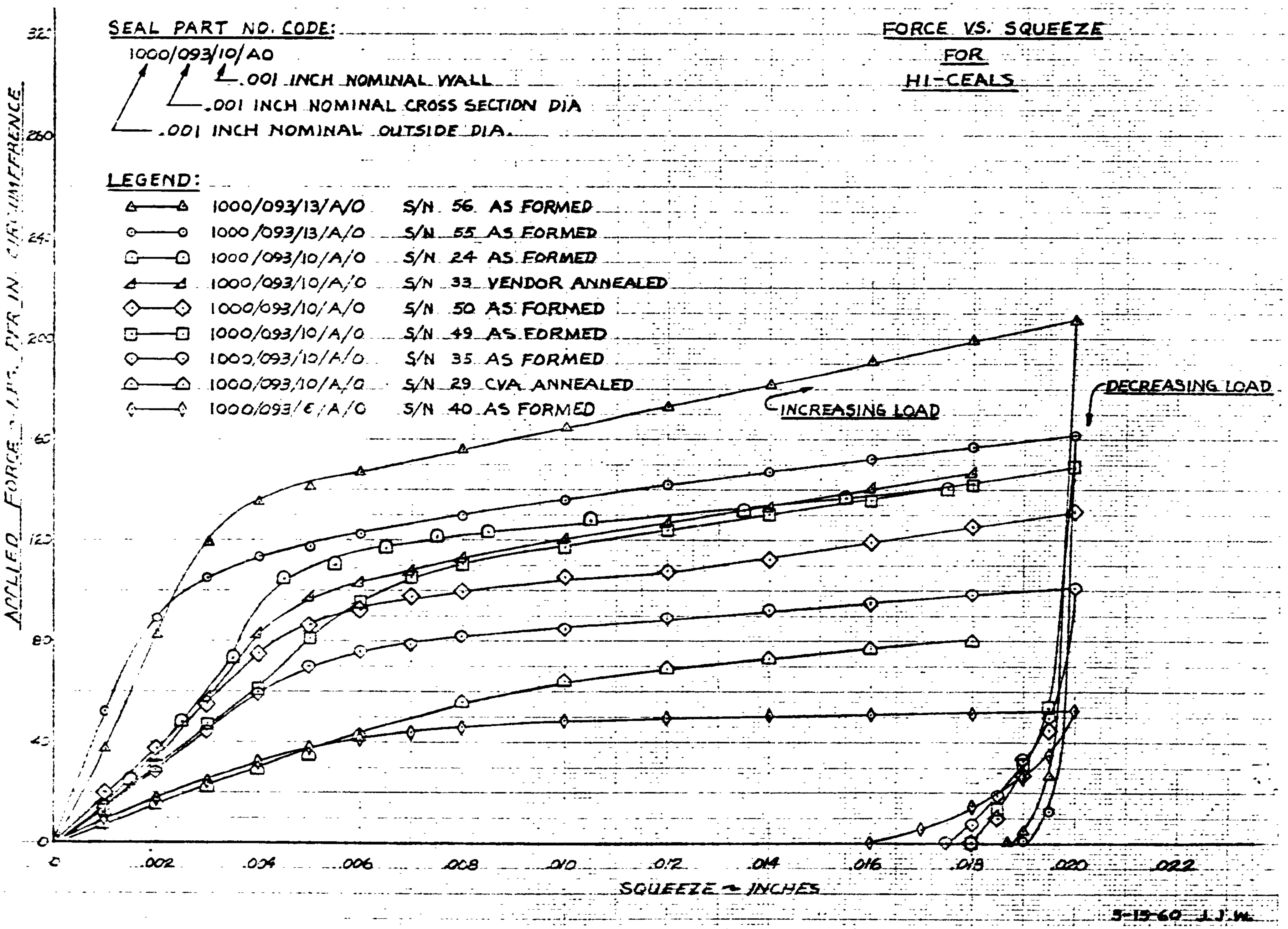
SEAL COMPRESSION TEST SETUP



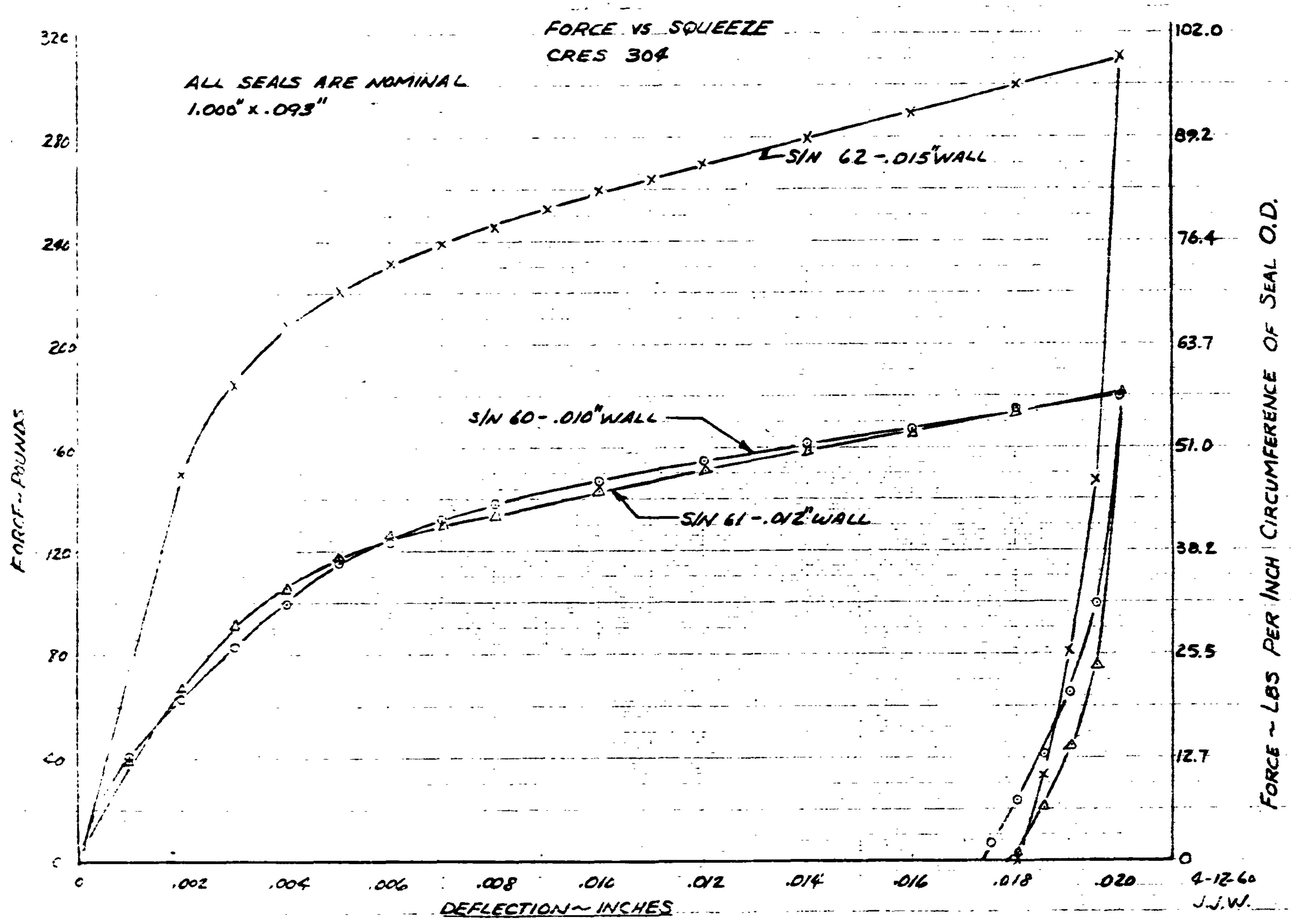
DETAIL A

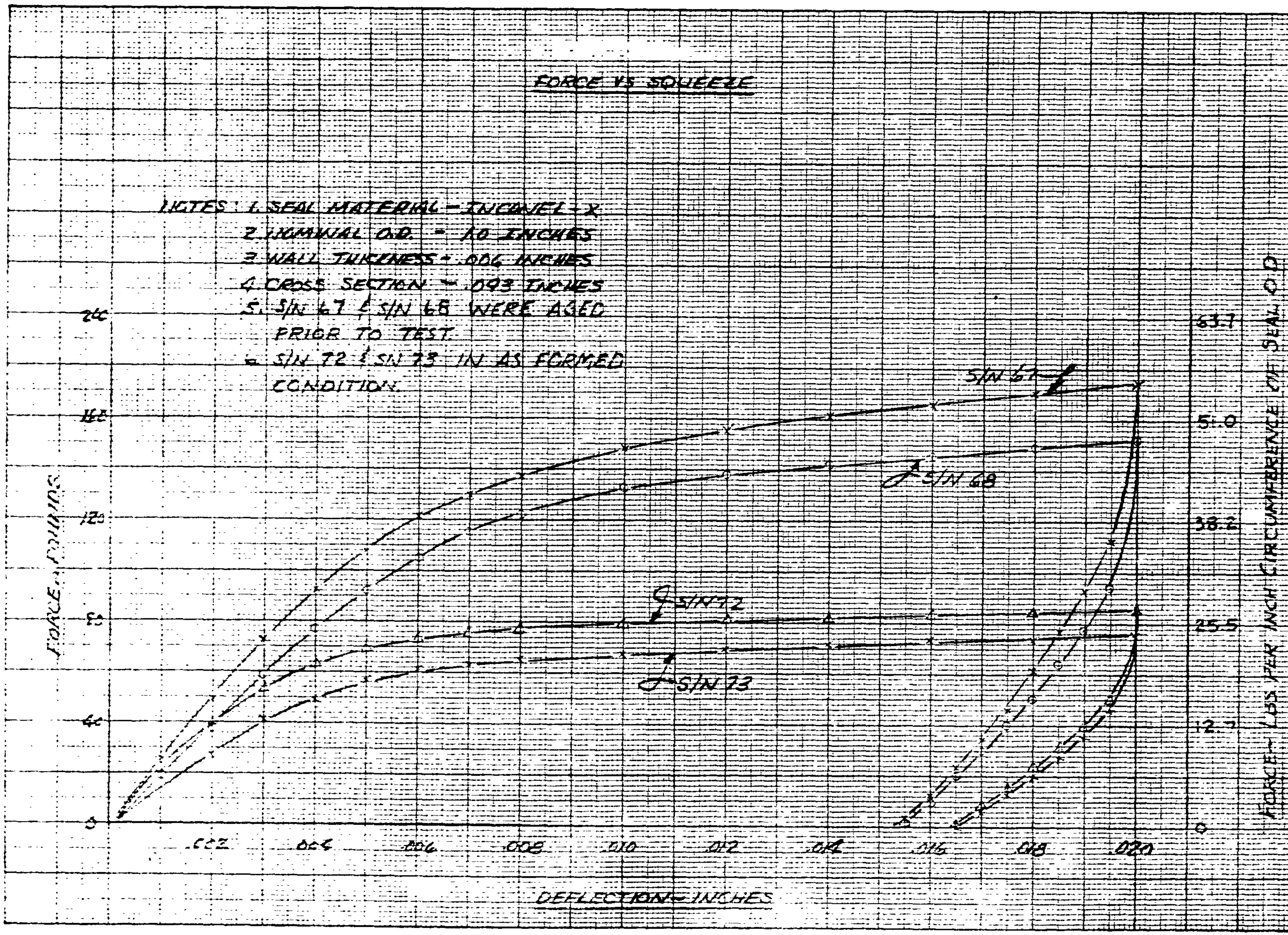


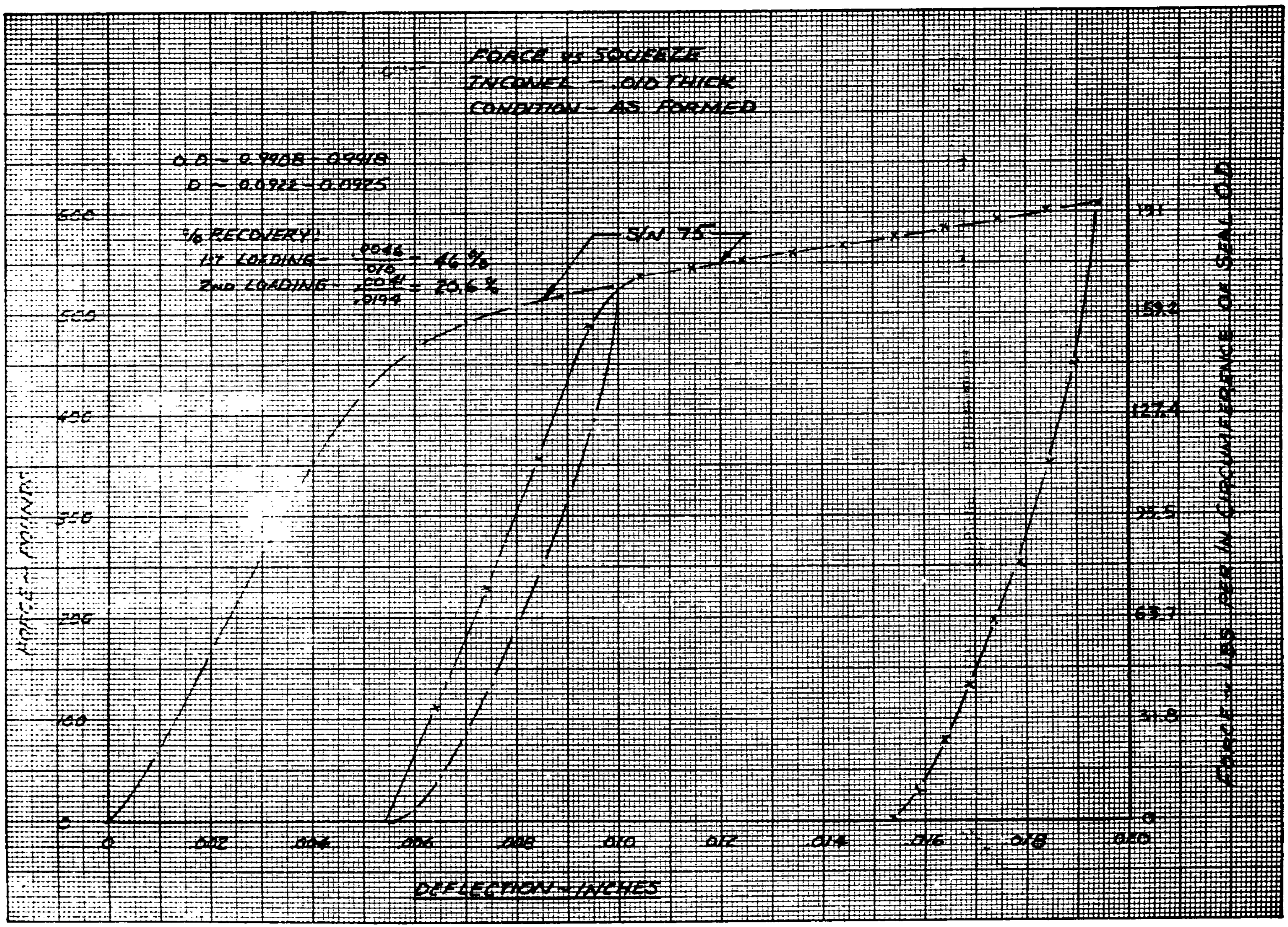
DRC

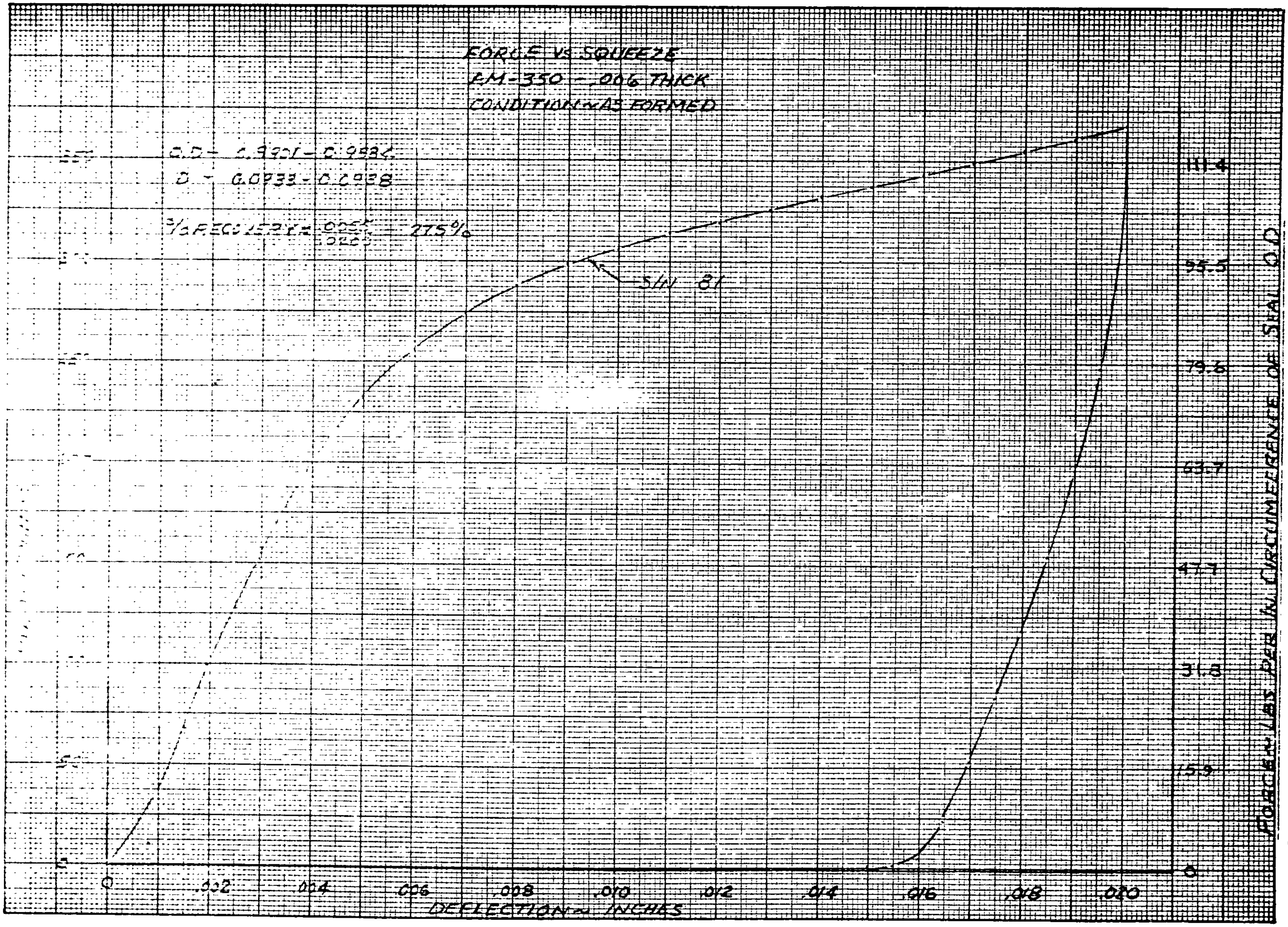


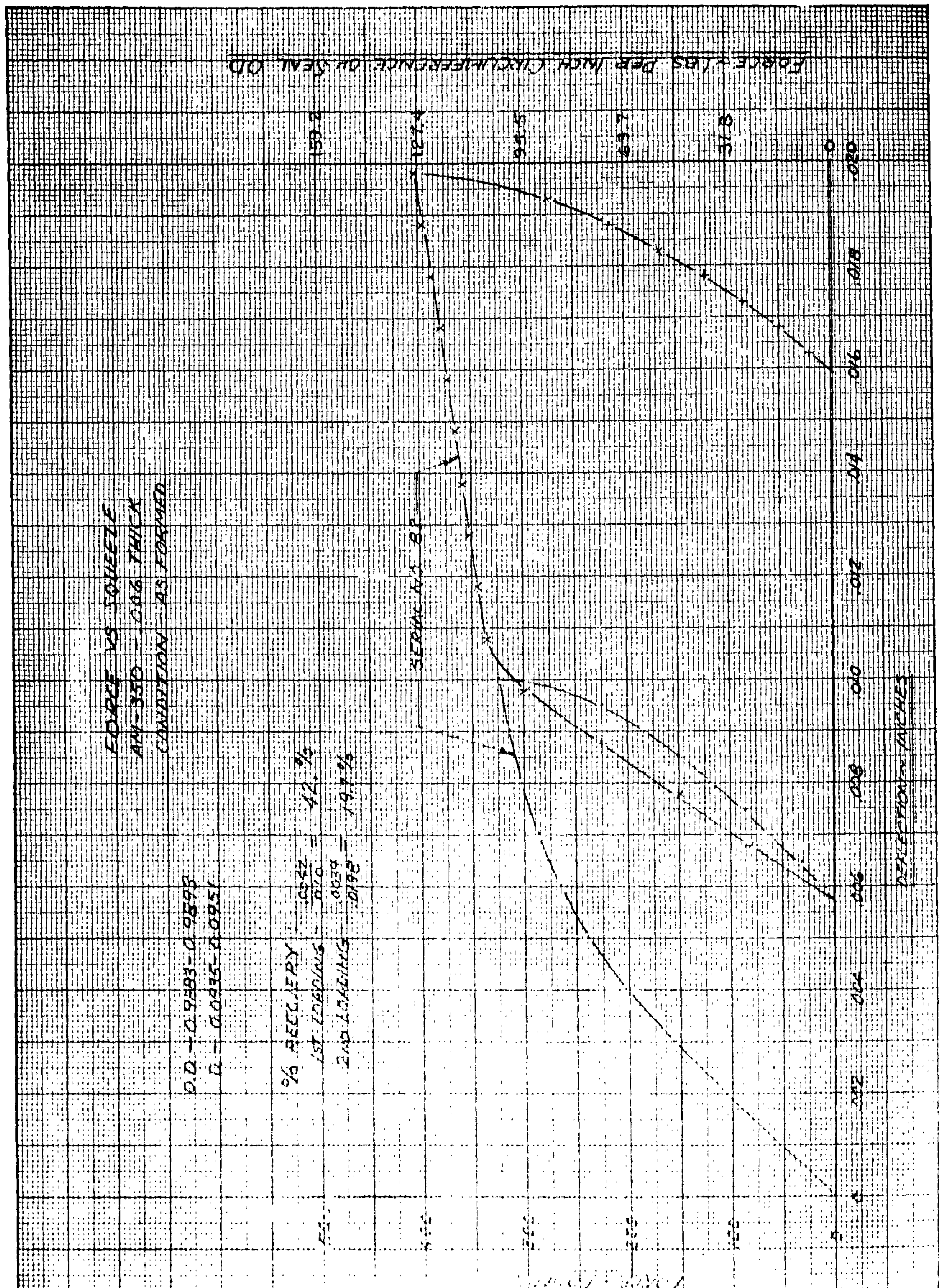
FORCE VS. SQUEEZE
FOR
HI-CEALS
SEAL PART NO. CODE:
1000/093/10/AO
.001 INCH NOMINAL WALL
.001 INCH NOMINAL CROSS SECTION DIA
.001 INCH NOMINAL OUTSIDE DIA.
LEGEND:
1000/093/13/A/O S/N 56 AS FORMED
1000/093/13/A/O S/N 55 AS FORMED
1000/093/10/A/O S/N 24 AS FORMED
1000/093/10/A/O S/N 33 VENDOR ANNEALED
1000/093/10/A/O S/N 50 AS FORMED
1000/093/10/A/O S/N 49 AS FORMED
1000/093/10/A/O S/N 35 AS FORMED
1000/093/10/A/O S/N 29 CVA ANNEALED
1000/093/6/A/O S/N 40 AS FORMED
INCREASING LOAD
DECREASING LOAD
APPLIED FORCE
0
40
80
120
200
280
0
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
.022
SQUEEZE ~ INCHES
5-15-60 J.J.W.

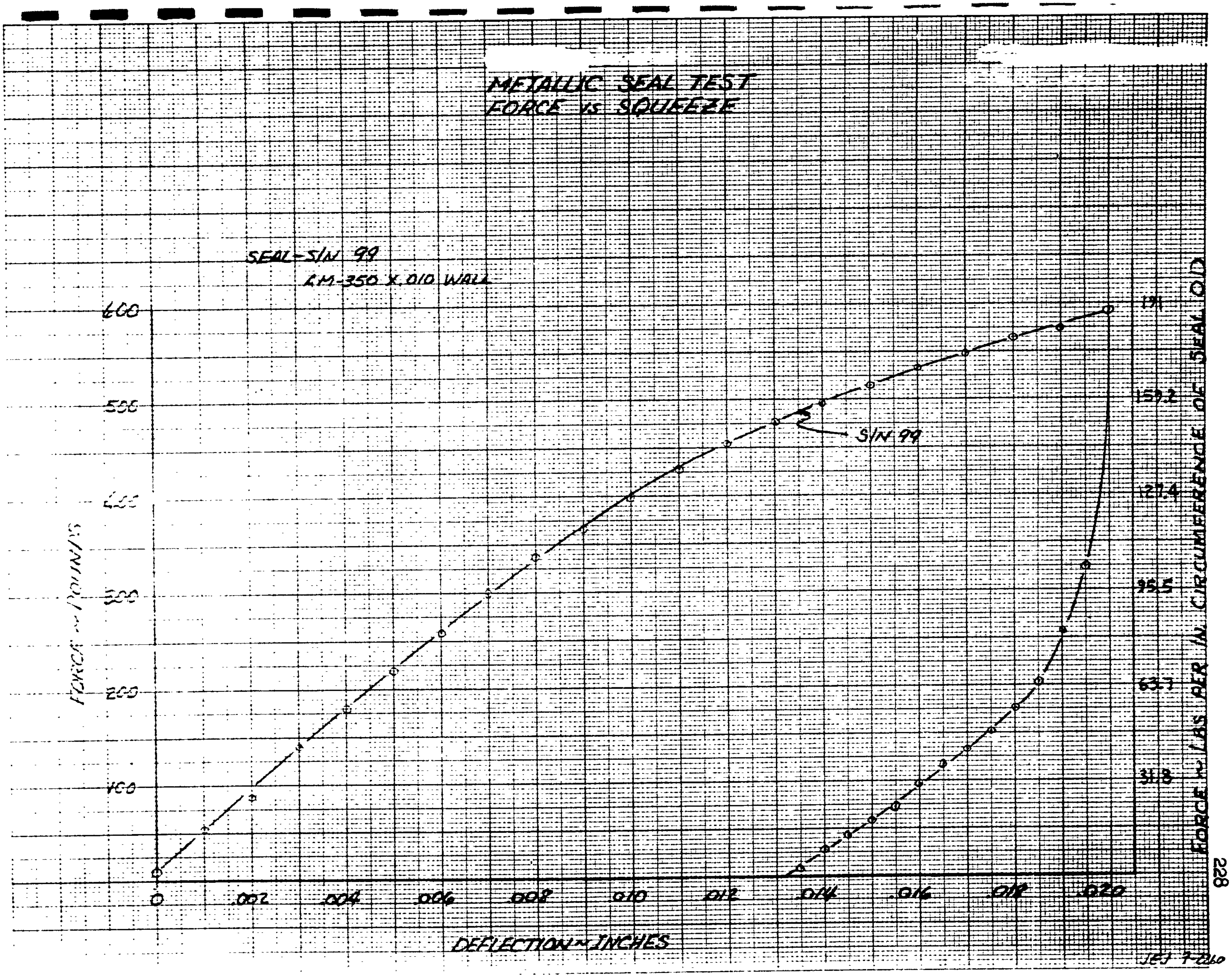

FORCE VS SQUEEZE
CRES 304
ALL SEALS ARE NOMINAL
1.000" x .093"
S/N 62 - .015" WALL
S/N 60 - .010" WALL
S/N 61 - .012" WALL
FORCE ~ POUNDS
FORCE ~ LBS PER INCH CIRCUMFERENCE OF SEAL O.D.
DEFLECTION ~ INCHES
4-12-60
J.J.W.

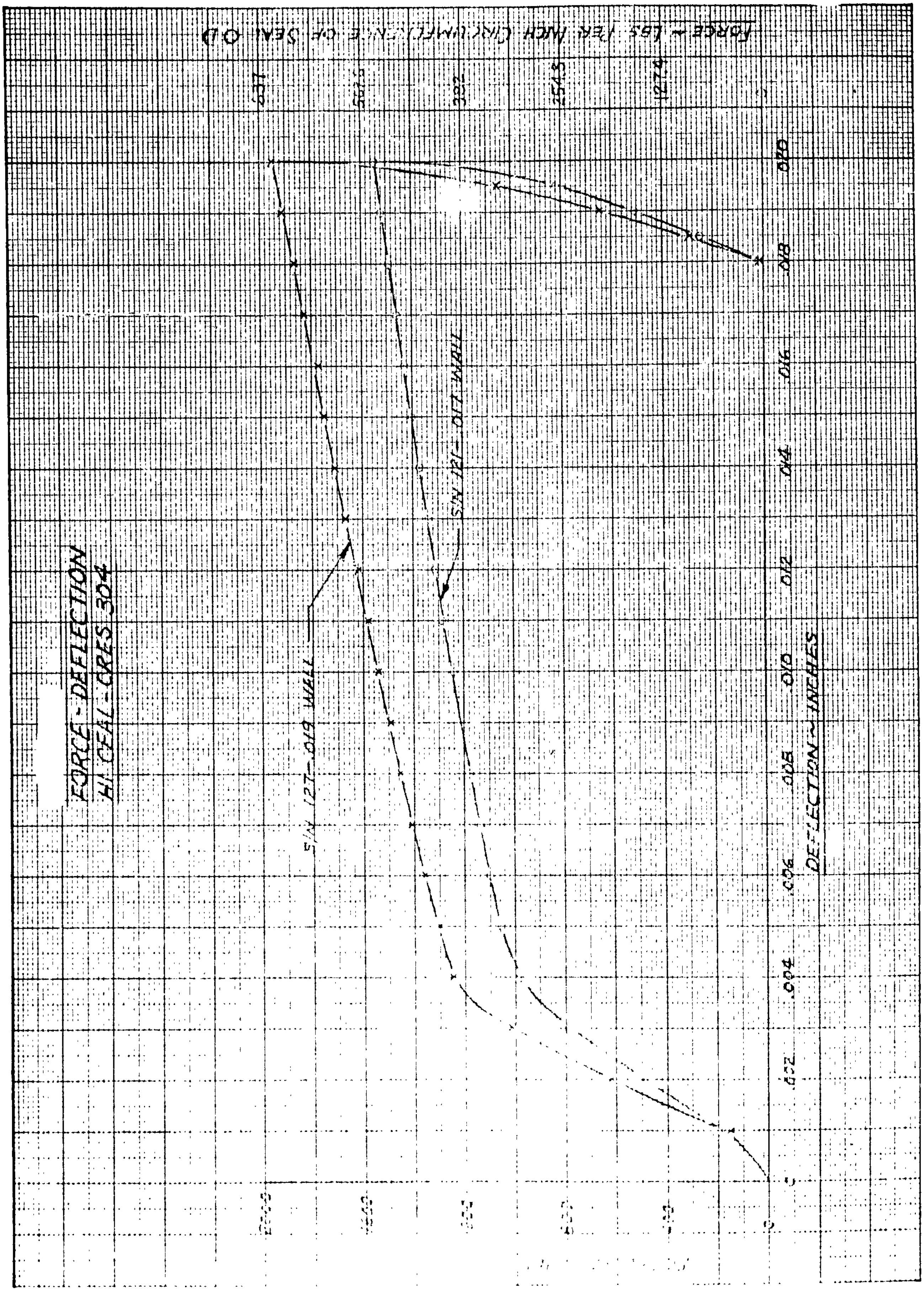

FORCE VS SQUEEZE
NOTES: 1. SEAL MATERIAL - INCONEL-X
2. NOMINAL O.D. - 1.0 INCHES
3. WALL THICKNESS - .006 INCHES
4. CROSS SECTION - .093 INCHES
5. S/N 67 & S/N 68 WERE AGED PRIOR TO TEST
6. S/N 72 & S/N 73 IN AS FORMED CONDITION
S/N 67
S/N 68
S/N 72
S/N 73
FORCE - POUNDS
0
40
80
120
160
200
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
DEFLECTION - INCHES
FORCE - LBS PER INCH CIRCUMFERENCE OF SEAL O.D.
0
12.7
25.5
38.2
51.0
63.7


FORCE vs SQUEEZE
INCONEL - .010 THICK
CONDITION - AS FORMED
O.D ~ 0.9908 - 0.9918
D ~ 0.0922 - 0.0925
% RECOVERY:
1ST LOADING - .0046 / .010 = 46 %
2ND LOADING - .004 / .0194 = 20.6 %
S/N 75
600
500
400
300
200
100
0
0
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
191
159.2
127.4
95.5
63.7
31.8
0
DEFLECTION - INCHES
FORCE ~ LBS PER IN CIRCUMFERENCE OF SEAL O.D
FORCE ~ POUNDS


FORCE VS SQUEEZE
RM-350 - .006 THICK
CONDITION ~ AS FORMED
S/N 81
111.4
95.5
79.6
63.7
47.7
31.8
15.9
0
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
DEFLECTION ~ INCHES
FORCE ~ LBS PER IN. CIRCUMFERENCE OF SEAL O.D.


FORCE VS SQUEEZE
AM-350 — .006 THICK
CONDITION — AS FORMED
O.D — 0.9883-0.9893
I.D — 0.0935-0.0951
% RECOVERY
1ST LOADING — .0042/.010 = 42. %
2ND LOADING — .0039/.0198 = 19.7 %
SERIAL NO. 82
FORCE ~ LBS PER INCH CIRCUMFERENCE OF SEAL OD
159.2
127.4
95.5
63.7
31.8
0
0
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
DEFLECTION ~ INCHES


METALLIC SEAL TEST
FORCE vs SQUEEZE
SEAL-S/N 99
AM-350 X .010 WALL
S/N 99
600
500
400
300
200
100
FORCE ~ POUNDS
0
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
DEFLECTION ~ INCHES
191
159.2
127.4
95.5
63.7
31.8
FORCE ~ LBS PER IN. CIRCUMFERENCE OF SEAL OD

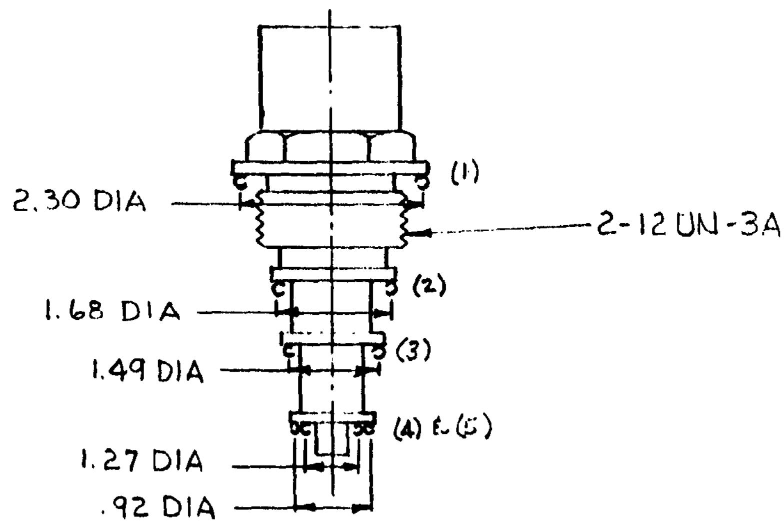

FORCE - DEFLECTION
HI CEAL - CRES 304
S/N 127 - .019 WALL
S/N 121 - .017 WALL
DEFLECTION ~ INCHES
0
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020

APPENDIX II-8

CALCULATION TO DETERMINE DESIRABLE LIMIT OF HI-CEAL COMPRESSIVE FORCE

APPENDIX II-7

CALCULATION TO DETERMINE DESIRABLE LIMIT OF HI-CEAL COMPRESSIVE FORCE



PRESSURE OPERATED SHUT-OFF VALVE
CLASS C

1. It is desirable to keep the compressive force of the Hi-Ceals as low as possible so that component installation torque will not be excessive. Because of the number and size of the Hi-Ceals involved, the Class C pressure operated shut-off valve will require the greatest installation torque. A maximum installation torque of 2,000 inch/pounds is estimated as a reasonable limit. This value should allow enough compressive force to effect low pressure sealing, while not causing unnecessary component installation torque problems.

2. Assume force to deflect Hi-Ceals is 100 pounds per inch of Hi-Ceal circumference.

3. Estimate coefficient of friction stainless steel on stainless steel to be 0.4 .

4. Radii to points of seal contact:

(a) 1.15 inches
(b) .84 inch
(c) .75 inch
(d) .63 inch
(e) .46 inch

5. **Torque caused by seal friction:**

$T_S = \mu P r$ μ = friction coefficient (.4)
P = load
r = radius

$T_{S_1} = .4\ (100\ \#/in.)(\pi\ 2.30\ in.)(1.15\ in.) = 332\ in.\ \#$

$T_{S_2} = .4\ (100\ \#/in.)(\pi\ 1.68\ in.)(.84\ in.) = 178$

$T_{S_3} = .4\ (100\ \#/in.)(\pi\ 1.49\ in.)(.75\ in.) = 141$

$T_{S_4} = .4\ (100\ \#/in.)(\pi\ 1.27\ in.)(.63\ in.) = 100$

$T_{S_5} = .4\ (100\ \#/in.)(\pi\ .92\ in.)(.48\ in.) = 56$

TOTAL (T_S) = 807 in. #

6. Torque caused by thread friction:

$$T_t = \gamma_t W \frac{\cos\theta \tan a + \mu_t}{\cos\theta - \mu_t \tan a}$$

where: T_t = torque

γ_t = thread pitch radius = $\frac{1.9459}{2}$ = .873

W = total load

θ = thread half angle (30°)

a = thread helix

$$\tan a = \frac{\text{Lead}}{\text{Pitch Circumference}} = \frac{.0833}{(1.9459)} = .0136$$

μ = coefficient of friction (.4)

$W = 100\ \#/in.\ \ (\pi \times 2.30) + (\pi \times 1.68) + (\pi \times 1.49) + (\pi \times 1.27) + (\pi \times .92)$

$W = 2400\ \#$

$$T_t = .873\ (2400)\ \frac{.866\ (.0136) + .4}{.866 - .4\ (.0136)}$$

$T_t = 1020\ in.\ \#$

7. Total Torque: $T = T_S + T_t$

$T = 807 + 1020$

$T = 1827\ in.\ \#$

7. Conclusions:

If 2,000 inch pounds is considered a maximum installation torque, Hi-Ceal compressive forces should not exceed 100 pounds per inch of seal circumference.

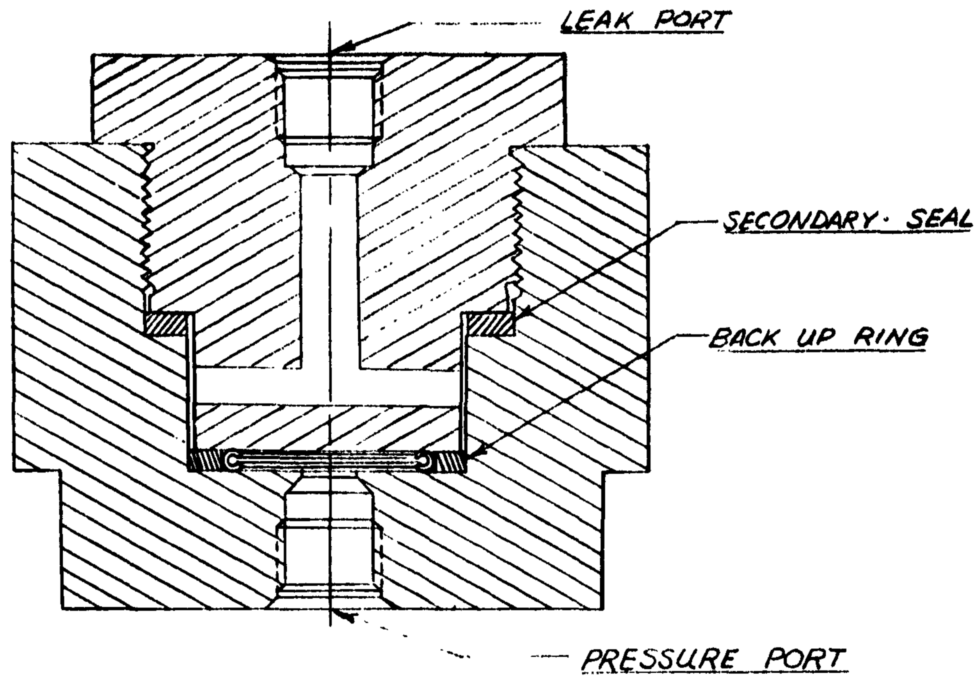
NOTE: It was later established that a higher seal compressive force was necessary and that excessive torque problems could be handled by presetting each seal prior to component installation (see discussion in Section 2).

APPENDIX II-9

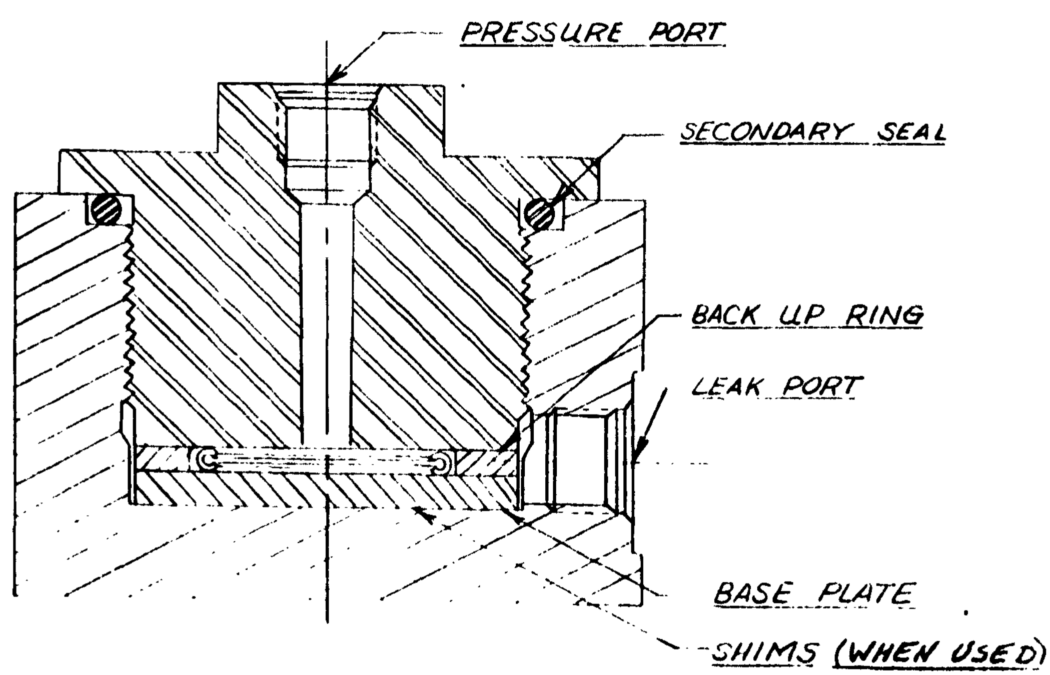
HI-CEAL TEST SUMMARY

TEST HOUSINGS
TEST SET UPS
TEST RESULTS

CVA HI-CEAL TEST HOUSINGS

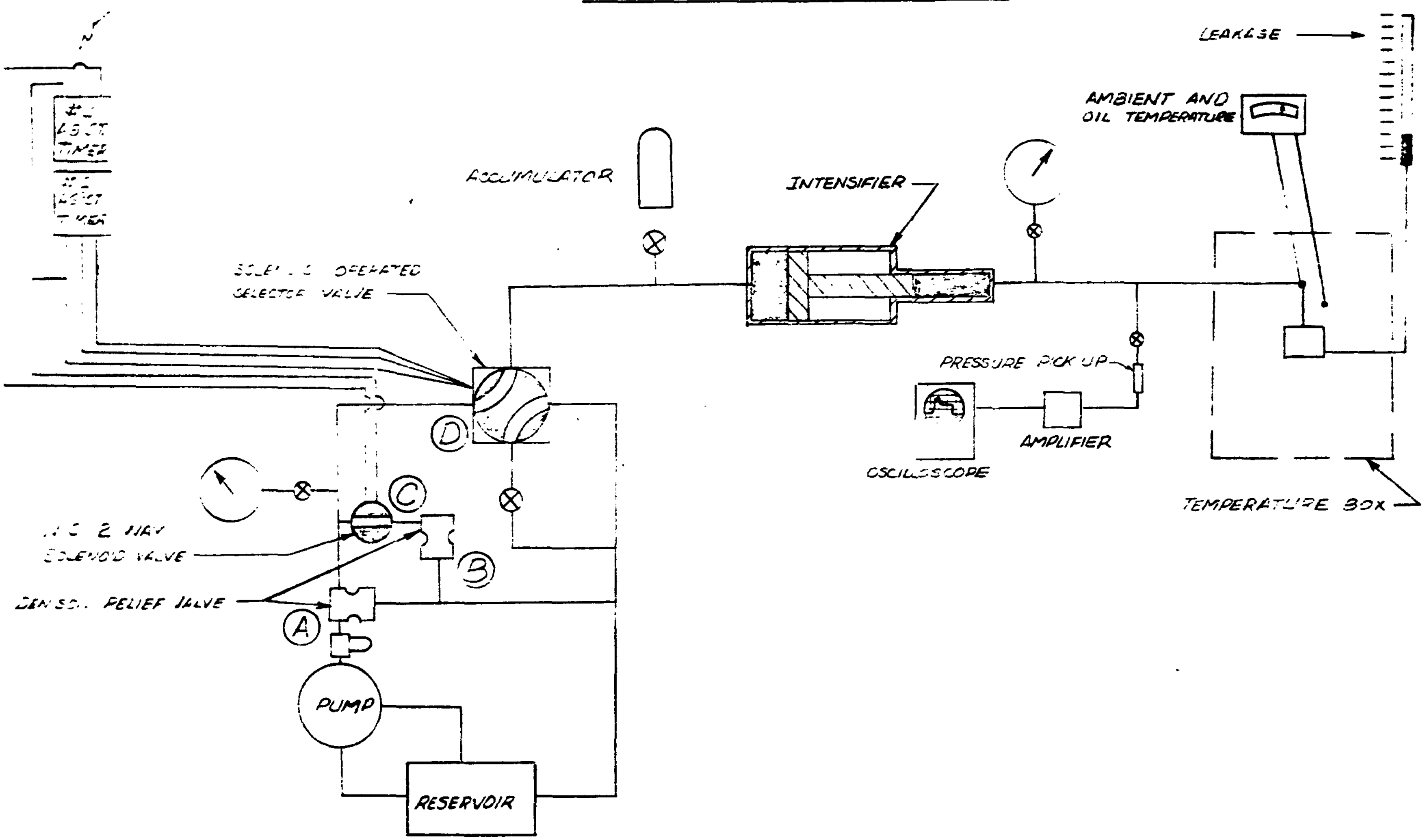


TL 3618 TEST HOUSING



TL 3628 TEST HOUSING

PRESSURE IMPULSE SET UP



REVISION DATE:

HI-CEAL TESTS

S/N	Seal Measured O.D.- In.	Depth-In.	Cavity Measured O.D.-In.	Depth-In.	Test Details
24	-	-	-	-	Compression tested in as formed condition - See Appendix II-7.
29	-	-	-	-	Compression tested after annealing and vapor honing See Appendix II-7.
33	-	-	-	-	Compression tested after annealing by vendor - See Appendix II-7.
25	.998-1.001	.088-.090	1.014	.087	Annealed and vapor honed by CVA. Vapor hone reduced depth from .092-.093 inches to .088-.090 inches. Seal leaked on application of internal pressure.
26	.998-1.001	.087-.092	1.014	.087	Annealed and vapor honed by CVA. Vapor hone reduced depth .092-.093 inches to .087-.092 inches. Seal leaked on application of internal pressure.
27	.999-1.000	.089-.092	1.014	.087	Annealed and vapor honed by CVA. Vapor hone reduced seal depth from .093-.095 inches to .089-.092 inches. Seal leaked on application of internal pressure.
28	.999-1.000	.093-.094			Annealed and cleaned with crocus cloth at CVA. Would not seal under any pressure to 300 psi. On removal seal found slightly dented. Seal free depth measured .0878-.0882 in., O.D. 1.002-1.003 inches. Reinstalled in .0790x1.000 cavity, slowly raised pressure to 6000 psi with no leakage. On removal, seal depth measured .0793-.0800 inches.

REVISION DATE

S/N	Seal Measured O.D.-In.	Seal Measured Depth-In.	Cavity Measured O.D.-In.	Cavity Measured Depth-In.	Test Details
34	.999-1.000	.093	1.006	.087	Annealed and cleaned with crocus cloth at CVA. Very minute leakage occurred on application of internal pressure.
31	.999-1.001	.092-.093	1.014	.087	Annealed and vapor honed at CVA. Slowly raised internal pressure to 1000 psi with small leakage occurring. Repeated several times with leakage diminishing each time. Leakage declined while holding pressure at 1000 psi for several minutes. Pressure cycled seal several times to 2000 psi very slowly. A pressure point of definite increased leakage occurred each time. On first cycle, leakage point was 1500±50 psi, and thereafter was 1800± 50 psi. Seal seemed to reseat at about same pressure. Removed seal. Cross section and O.D. measured .0880-.0885 and 1.0070-1.0125 inches respectively. Installed again in same cavity, slowly raised pressure to 6000 psi. Significant leakage occurred between 2000 and 4500 psi but stopped completely by time 6000 psi was reached. Removed seal and measure depth and O.D. to be .0880-.0885 and 1.014 inches respectively. Installed seal again in same cavity. Internal hydraulic static pressure tests to 6000 psi. revealed slight leakage between 1000 and 5000 psi with zero leakage at 6000 psi. Stabilized seal and test housing at temp. of 450 deg. F. and found seal to leak excessively at low pressure. After cooling and stabilizing at temp. of 80±20 deg. F., slowly applied internal pressure to 6000 psi with only slight leakage at intermediate pressures and zero leakage at 6000 psi. Stabilized seal and housing a second time at a temp. of 450 deg. F., seal again leaked excessively at low pressure.

REVISION DATE:

S/N	Seal Measured O.D.-In.	Seal Measured Depth-In.	Cavity Measured O.D.-In.	Cavity Measured Depth-In.	Test Details
32	.999-1.000	.092-.093	1.014	.087	Annealed and vapor honed by CVA. Installed and removed without application of pressure to check permanent set. Seal measured .088x1.002-1.003 inches upon removal.
30			1.014	.087	Annealed and vapor honed by CVA. Normal (internal) pressure slowly raised to 3000 psi with slight leakage resulting. Dropped internal pressure to zero then applied reverse pressure to seal. Seal collapsed at 2400 psi.
36	.988-.989	.095	1.001	.091	Installed and slowly applied proof press. at room temp. to 6000 psi. There was no leakage apparent. After soaking seal and housing at 450°F for 1 hour, leakage occurred on application of pressure. Removed seal and measured dimensions to be .992-.993x.092-.093 in. Re-installed seal and slowly applied press. to 6000 psi without leakage. Upon raising temp. and soaking for 30 min. at 450°F, seal leaked at pressures above 500 psi. After cooling overnight, seal held without leakage to 6000 psi. Initial installation torque for seal was 270 in-lb.
37	.988	.095-.096	1.001	.083	Installed and slowly applied proof press. at room temp. to 6000 psi. There was no leakage. After soaking seal and housing at 450°F for 1 hour, leakage occurred on application of pressure. After seal and housing cooled to room temp., no leakage occurred on application of press. to 6000 psi. Removed and measured dimensions to be .998-.999x.0845 in.

FORM NO. 38

REVISION DATE:

S/N	Seal Measured O.D.-In.	Seal Measured Depth-In.	Cavity Measured O.D.-In.	Cavity Measured Depth-In.	Test Details
38	.987-.991	.095-.096	1.001	.075	Installed and slowly applied proof press. at room temp. to 6000 psi there was no leakage. After soaking seal and housing at 450°F for 1 hour, pressure was slowly applied to 6000 psi, without leakage occurring. Removed seal and measured dimensions to be 1.000x.007 in. Materials Lab checked hardness to be Rc25.
40	-	.093-.094	-	-	Ran comp. test. See Appendix II-7. After test seal depth was measured to be .077-.079 in.
35	-	.094-.095	-	-	Ran comp. test. See Appendix II-7. After test seal depth was measured to be .078-.079 in.
49	-	.094	-	-	Ran comp. test. See Appendix II-7 test seal depth was measured to be .076 in.
50	-	.094	-	-	Ran comp. test. See Appendix II-7, After test seal depth was measured to be .076 in.
41	.988	.094	1.000	.088	Installed in test housing furnished by vendor. Slowly applied proof press at room temp. to 6000 psi. No leakage occurred. Soaked seal and housing at 450°F for 1 hour. On application of press., pipe threads in housing began leaking. Had to remove from oven. The seal and housing were later installed in the oven and soaked at 450°F for 1 hour, pressure was gradually increased to 6000 psi at this temp. without evidence of leakage.
42	.990	.095	1.001	.091	Installed and slowly applied proof press at room temp. to 6000 psi with no leakage occurring. Soaked seal and housing for 1 hour at 450°F slowly applied proof press to 6000 psi. Approx. 2 cc of fluid was lost by leakage around 2500 psi. No leakage occurred in subsequent press applications and during approx. 1000 press impulse cycles at 450°F.

. . . ON DATE:

S/N	Seal Measured O.D.-In.	Seal Measured Depth-In.	Cavity Measured O.D.-In.	Cavity Measured Depth-In.	Test Details
43	.989	.095	1.001	.083	Installed and slowly applied proof press. at room temp. to 6000 psi with no leakage occurring. Soaked seal and housing for 1 hour at 450°F. Slowly applied proof press. to 6000 psi with no evidence of leakage, however, small amounts of leakage could not be detected because of loss of fluid from leak detector lines. Applied approx. 1000 press. impulse cycles with no evidence of leakage at 450°F.
51	.989	.094	.990	.081	Installed and slowly applied proof press. at room temp. to 6000 psi with no leakage occurring. Repeated this procedure after 1 hour soak at 450°F. Removed seal after 5,700 pressure impulse cycles and found visual indication that seal was beginning to crack in a plane perpendicular to its axis along the outer periphery which first contacted the cavity O.D.
52	.989	.095	.990	.089	Same test results as S/N 51 except seal had begun to leak after 5,700 p.i.c. and seal had split along outer periphery
53	.989	.095	1.001	.083	Same test results as S/N 52 seal
54	.986-.989	.096	1.000	.073	Squeezed in cavity, removed and measured to be 1.002 x .076 in. No pressure was applied. Torque required for installation was 60 ft-lb.
57	.990-.991	.090	1.000	.088	Installed seal in cavity and slowly raised pressure to internal diameter of seal to 6000 psi at room temp. without any leakage occurring. Soaked seal and housing at 450°F for one hour and then applied internal hydraulic pressure. After cooling overnight to room temp. seal held pressure to 6000 psi without leakage. Removed and measured seal depth and O.D. to be .0885-.0895 and .9995 inches respectively.

PAGE NO.

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REVISION DATE:

S/N	Seal Measured O.D.-In.	Seal Measured Depth-In.	Cavity Measured O.D.-In.	Cavity Measured Depth-In.	Test Details
57*	.9995	.0885-.0895	1.000	.083	Installed seal in cavity and slowly raised pressure applied to internal diameter of seal to 6000 psi at room temp. and again after soaking for one hour at 450°F without any leakage occurring. Seal impulse pressure cycled to 6000 psi at temperature from -65°F to 450°F for a total of 49,000 cycles. Seal leaked during impulse cycling as much as 40 cc per hour. Upon removal, found seal split around outer periphery into two equal halves.
58	.990	.090	1.000	.088	Same test and results as obtained for S/N 57, except seal depth and O.D. was .0898-.0902 and .998 inches respectively after removal.
58*	.998	.0898-.0902	1.000	.079	Same test and results as S/N 57 (including splitting of seal) on second installation except leakage rate during impulse cycling was approximately one half as great as S/N 57.

* Indicates second installation

SUMMARY OF HI-CEAL HYDRAULIC TESTS
FOR MARCH, APRIL, MAY 1960

S/N	Seal Measured O.D. - In.	Seal Measured Depth - In.	Cavity Measured O.D. - In.	Cavity Measured Depth - In.	Test Details
60	1.000 (Nominal)	.093	None		CRES 304, .010 wall thickness, as formed conditio Compression tested. See Appendix II-7 for test results.
61	1.000 (Nominal)	.093	None		CRES 304, .012 wall thickness, as formed condition. Compression tested.
62	1.000 (Nominal)	.093	None		CRES 304, .015 wall thickness, as formed condition. Compression tested.
63	1.000 (Nominal)	.093	1.000	.0790	CRES 304, .015 wall thickness, as formed condition. Successfully sustained proof pressure test. 37,560 impulse cycles were applied at room temperature. No leakage occurred.
64	1.000 (Nominal)	.093	1.00	.0887	CRES 304, .015 wall thickness,as formed condition. Successfully sustained proof pressure test. 37,560 impulse cycles were applied at room temperature. No leakage occurred.
65 (aged)	.990-1.006	.0905-.0928	1.0145	.087	Inconel-X installed seal in cavity and slowly raised pressure applied to internal diameter of seal to 6000 psi at room temp. without any leakage occurring. Soaked seal and housing at 450°F for one hour. Excessive leakage occurred at low pressure. After cooling to room temp. the seal held pressure to 6000 psi without leakage. Removed seal from the cavity and measured. Then installed in second cavity. Slowly raised pressure to internal diameter of seal to 6000 psi at room temp. without any leakage occurring. Began impulse pressure cycling at 450°F. Leakage was excessive after 3750 cycles. Removed seal from housing and measured; O.D. 1.011-1.012 in., depth .0875-.0880 in. Surfaces of the housing which contacted the seal showed evidence of wear
65* (aged)	1.001-1.011	.0882-.0902	1.014	.0845	

SUMMARY OF HI-CEAL HYDRAULIC TESTS FOR MARCH, APRIL, MAY 1960

S/N	Seal Measured O.D. - In.	Seal Measured Depth - In.	Cavity Measured O.D. - In.	Cavity Measured Depth - In.	Test Details
66 (aged)	.991-1.008	.0905-.0920	1.0142	.074	Inconel-X-Installed seal in cavity and slowly raised pressure to the internal diameter of seal to 6000 psi at room temp. and again after soaking for one hour at 450°F without any leakage occurring. Began impulse pressure cycline at 450°F. Leakage was 1.0 cc/min after 5915 cycles. Removed seal from housing and measured; O.D. 1.0125-1.0130 in., Depth .0775-.0780 In. Surfaces of the housing which contacted the seal showed evidence of wear.
67	1.000 (Nominal)	.093		None	Inconel-X, .006 wall, thickness. Compression tested. See App.II-7 for test results. This seal was aged prior to testing.
68	1.000 (Nominal)	.093		None	Inconel-X, .006 wall thickness. Compression tested. See App.II-7 for test results. This seal was aged prior to testing.
69	.999-1.000	.0908-.0913	.999	.083	Inconel-X, Installed seal in cavity and slowly raised pressure to internal diameter of seal to 6000 psi at room temp. and again after soaking for 1.5 hours at 450°F without any leakage occurring. Began impulse cycling at 450°F. Leakage was .5 cc/min after 7515 cycles. Removed seal from housing and measured; O.D. 1.0001 in., depth .0870 - .0875 In. Surfaces of the housing which contacted the seal showed evidence of wear.
70 (aged)	.999-1.000	.0895-.0923	1.001	.073	Installed seal in cavity and slowly raised pressure to the internal diameter of seal to 6000 psi at room temp. and again after soaking for 1.5 hours at 450°F without any leakage occurring. Began impulse cycling at 450°F. Leakage was 1 cc/min after 7515 cycles. Removed seal from housing and measured; depth .0782-.0789 in. The seal was tight in the cavity. Surfaces of the housing which contacted the seal showed evidence of wear.

SUMMARY OF HI-CEAL HYDRAULIC TESTS
FOR MARCH, APRIL MAY 1960

S/N	Seal Measured O.D. - In.	Seal Measured Depth - In.	Cavity Measured O.D. - In.	Cavity Measured Depth - In.	Test Details
72	(Nominal) 1.000	.093		None	Inconel-X, .006 wall thickness, as formed condition. Compression tested. See Figure V-2 for test results.
73	(Nominal) 1.000	.093		None	Inconel-X, .006 wall thickness, as formed condition. Compression tested. See Figure V-2 for test results.
74 (as formed)	.980	.0929-.0931	1.000	.0885	Installed seal in cavity and slowly raised pressure to internal diameter of seal to 6000 psi at room temp. and again after soaking at 450°F for .5 hours. No leakage occurred at room temp. Seepage occurred, at 450°F. Leakage was 1 cc/min after 4540 cycles.. Removed seal from cavity and measured O.D. .999-1.000 in., depth .0916-.0925 In. Surfaces of the housing which contacted the seal showed evidence of wear.
75	(Initial) .9908-.9918 (Post Test) 1.0100-1.0101	.0922-.0925 .0598-.0612		None	Inconel-X, .010 wall thickness, as formed condition. This seal was compression tested without a backup ring at room temperature. Figure V-3 presents the results of this test.
76	(Initial) .9895-.9922 (Post Test #1) .9928-.9930 (Post Test #2) .9930-.9945	.0900-.0910 .0870-.0870 .0855-.0858	 .9930 .9930	 (Test #1) .0840 (Test #2) .0822	Inconel X, .010 wall thickness, as formed condition. Chrome plated base was used for wear comparison. Sustained proof pressure at room temperature satisfactorily. Leaked excessively at all pressures at 450°F. After cooling over-night, 450°F was again applied. Leakage was slight at all pressures. Pressure was reduced. Leakage was excessive at low pressures (less than 1000 psi; Twenty impulse cycles were applied to 6000 psi. Pressure application after 20 cycles revealed excessive low pressure leakage. Removed seal to increase squeeze. Leakage again unacceptable throughout the test. Cycled for 11575 cycles. Chrome base looks good for wear resistance. Fretting corrosion evident on 17-f PH plug.

SUMMARY OF HI-CEAL HYDRAULIC TESTS
FOR MARCH, APRIL MAY 1960

S/N	Seal Measured O.D. - In.	Seal Measured Depth - In.	Cavity Measured O.D. - In.	Cavity Measured Depth - In.	Test Details
77	(Initial) .9882-.9889 (Post Test) .9905-.9910	.0928-.0952 .0902-.0905	1.0145	.0858	Inconel-X, .010 wall thickness, as formed condition. Installed in test housing to see if wear ring was initiated by tightening during installation. Wear ring was evident on both plug and seat after removal from test housing. Hardness test-plug C-40, seat C-35, Seal C-29.
79	(Initial) .9908-.9912 (Post Test) .9920-.9925	.0940-.0950 .0912-.0915	.9930	.0885	Inconel-X, .010 wall thickness, as formed condition. Satisfactorily sustained proof pressur at room temperature. Leaked badly at low pressure at 450°F. No cycles were attempted on this seal. On removal and inspection, the plug was found to have two spots where wear marks were absent. Leakage apparently was from these areas.
81	(Initial) .9834-.9901 (Post Test) .9991-1.0005	.0933-.0938 .0776-.0789		None	AM-350, .006 wall thickness, as formed condition. This seal was compression tested at room temperature without a backup ring. App.II-7 presents the results of this test.
82	(Initial) .9883-.9893	.0935-.0951		None	AM-350, .006 wall thickness, as formed condition. This seal was compression tested at room temperature without a backup ring.
83	(Initial) .987-.989 (Post Test) .999-.9988	.0923-.0932 .0780-.0785	1.0000	.075	AM-350, .006 wall thickness, as formed condition. Satisfactorily sustained proof pressure at room temperature and at 450°F. Started leaking excessively at 9532 cycles. Continued cycling to 15,115 cycles. Seal was found to be cracked open approximately half way around. Unopen crack around the other half. 17-4 PH plug and base had fretting corrosion wear marks.

SUMMARY OF HI-CEAL HYDRAULIC TESTS
FOR MARCH, APRIL MAY 1960

S/N	Seal Measured O.D. - In.	Seal Measured Depth - In.	Cavity Measured O.D. - In.	Cavity Measured Depth - In.	Test Details
84	(Initial) .9885-.9895 (Post Test) .9990-.9993	.0942-.0968 .0908-.0915	1.0000	.089	AM-350, .006 wall thickness, as formed condition. Satisfactorily sustained proof pressure at room temperature. Leaked badly at low pressures at 450°F. After cooling overnight, proof pressure test was done again at 450°F. This time there was zero leakage at all pressures. Seal leaked slightly from beginning of impulse cycling and became worse at 4555 cycles. Cycling was stopped at 10,737 cycles. Hi-ceal was split in half, fretting corrosion evident on plug and base plate.
86	(Initial) .9879-.9897 (Post Test) .9880-.9892	.0923-.0933 .0842-.0904	.9908	.0868	AM-350, .006 wall thickness, as formed condition. Satisfactorily sustained proof pressure test at room temperature and 450°F. Started gradual leaking at 4000 cycles. Leakage was fairly constant at approximately .3 cc/min. to 9866 cycles. Leakage then increased to approximately .- cc/min. cycling was stopped at 12,994 cycles. Ring, base plate, and plug had evidence of fretting corrosion.
87	-	-	-		Chromized this seal. During processing the seal was placed on end in a retort and heated to 1900°F. The seal sagged out of round and was fractured when an effort was made to straighten it.
91	(Initial) .9985-.9993 (Post Test) .9985-1.0003	.0787-.0841 .0733-.0756	1.0095	.046	15-7 Brazed Double "X" seal. .010 material thickness. Compression tested in a cavity at room temperature. Force required to squeeze was excessive. App.II-7 presents the results of this test.

SUMMARY OF HI-CEAL HYDRAULIC TESTS FOR MARCH, APRIL MAY 1960

S/N	Seal Measured O.D. - In.	Seal Measured Depth - In.	Cavity Measured O.D. - In.	Cavity Measured Depth - In.	Test Details
92	(Initial) .9993-.9999 (Post Test) 1.0008-1.0012	.0741-.0798 .0658-.0701	1.0095	.046	15-7 brazed double "X" seal. .010 material thickness. Compression tested in a cavity at room temperature. Force required to squeeze was excessive. App.II-7 presents the results of this test.
93	(Initial) .9880-.9890 (Post Test) .9901-.9902	.0950-.0962 .0891-.0898	.9908	.0868	Type 304, .015 wall thickness, as formed condition. Satisfactorily sustained proof pressure test at room temperature and 450°F. Leakage for this seal was slight. Total cycles = 14896. Wear marks on plug, seal, and base plate appear to be as on the 350 and Inconel-X. However, the scrubbing action on the plug and base plate was reduced, i.e., A thinner line of wear, with fretting corrosion marks, is evident. This may be explained by the increased stiffness due to a thicker wall.
94	(Initial) .9880-.9900 (Post Test) .9908-.9913	.0843-.0872 .0795-.0799	.9900	.0782	Type 304, .015 wall thickness. As formed condition ground .04 inch flats each side of seal. Satisfactorily sustained proof pressure test at room temperature. Leaked excessively at all pressures at 450°F. Started impulse cycles to evaluate wear on the plug and base plate. The bearing line on the seal fell outside the ground flats and scribed a very thin wear ring in plug and base plate. Total cycles - 9177.

SUMMARY OF HI-CEAL HYDRAULIC TESTS PERFORMED AT CHANCE VOUGHT AIRCRAFT, INCORPORATED FOR JUNE, JULY, AUGUST 1960

S/N and Seal	Seal Measurement O.D.-(In.)	Seal Measurement Depth-(In.)	Cavity Measurement I.D.-(In.)	Cavity Measurement Depth-(In.)	Test Details
85 AM 350 .006 wall as formed	(Initial) .990-.9910	.0948-.0970	.9930 (17-4 plug, 440-c base) (Finish 10 rms) (Hardness-Rc 35)	.0893	Sustained proof pressure. Impulse cycled 21,750 times. 17-4 plug and side of seal facing 17-4 plug had considerable number of fretting corrosion pits. 440-c base plate had wear line but little corrosion. Leakage rates: @3000 cycles, 2 cc/hr.; @6000 cycles, 2 cc/hr.; @9000 cycles, 5 cc/hr.; @12,000 cycles, 8 cc/hr., @15,000 cycles, 10 cc/hr.; @18,000 cycles, 19 cc/hr.; @21,750 cycles, 28 cc/hr.
	(Post Test) .9905-.9912	.0925-.0935			
	(Hardness-Rc 28)				
	Squeeze: .0055 min.; .0077 max.				
	Diametral clearance: .0020 min.; .0030 max.				
	First Installation				
88 AM 350 .006 wall as formed	(Initial) .9825-.9950	.0940-.0965	.9930 (17-4 plug and base) (Finish 12 rms) (Hardness-Rc 35)	.0868	Sustained proof pressure but leaked under impulsing. Fretting corrosion evident on all sealing faces after 8840 cycles.
	(Post Test) .9924-.9940	.0895-.0901			
	Squeeze: .0072 min.; .0097 max.				
	Diametral Clearance: .0080 min.; .0105 max.				
	Second Installation				
	(Initial) .9924-.9940	.0895-.0901	.9990 (Chrome plated plug & base) (Finish 8 rms) (Hardness-Rc 60+)	.0830	Sustained proof pressure at room temperature but leaked badly at 450°F proof pressure test. Impulse cycled 3000 times for seat wear evaluation. Plug and base in very good condition. Wear ring was just a thin line
	(Post Test) .9985-.9995	.0895-.0858			
	*Squeeze: .0110 min.; .0135 max.				
	*Diametral Clearance: .0040 min.; .0165 max.				
	*Computed from initial seal dimensions				

SUMMARY OF HI-CEAL HYDRAULIC TESTS PERFORMED AT CHANCE VOUGHT AIRCRAFT, INCORPORATED FOR JUNE, JULY, AUGUST 1960

S/N and Seal	Seal Measurement O.D.-(In.)	Seal Measurement Depth-(In.)	Cavity Measurement I.D.-(In.)	Cavity Measurement Depth-(In.)	Test Details
89 AM 350 .006 wall as formed	(Initial) .9899-.9902 (Post Test) .9940-.9942	 .0935-.0949 .0885-.0891	.9980 (17-4 plug, 416 base) (Finish 12 rms) (Hardness-Rc 35)	.0855	Sustained proof pressure leakage tests. Impulse cycled 14,485 times. 17-4 plug and side of seal facing 17-4 plug had fretting corrosion pits. 416 base plate had little corrosion but more than 440c had (ref S/N 85). Leakage rates: @3000 cycles, 2.5 cc/hr.; @6000 cycles, 2.5 cc/hr.; @9000 cycles, 3.5 cc/hr.; @12,000 cycles, 17 cc/hr.; @14,485 cycles, 24 cc/hr.
	Squeeze: .0080 min.; .0094 max. Diametral Clearance: .0078 min.; .0081 max.				
95 CRES 304 .015 wall Electroless Nickel Plate	(Initial) 1.0059-1.0070 (Post Test) No Measurement	 .0932-.0939	1.0145 (17-4 plug and base) (Finish 12 rms) (Hardness-Rc 35)	.0870	Installed in cavity and then removed to check scoring of seats on installation. Nickel plate was found to be flaked off of seal. No pressure testing was done.
	Squeeze: .0062 min.; .0069 max. Diametral Clearance: .0075 min.; .0086 max.				
96 CRES 304 .015 wall as formed	(Initial) 1.0015-1.0026 (Post Test) 1.0055-1.0060	 .0892-.0895 .0836-.0842	1.0060 (Chrome plated plug & base) (Finish 8 rms) (Hardness-Rc 60+)	.0815	Sustained proof pressures at room temperature and 450°F. Started leaking badly at 3840 impulse cycles. Stopped test after 5600 cycles. Very little wear on plug and base. Leakage rates: @3000 cycles, zero; @4000 cycles, 18 cc/hr.; @5600 cycles, 39.3 cc/hr.
	Squeeze: .0077 min.; .0080 max. Diametral Clearance: .0034 min.; .0045 max.				

SUMMARY OF HI-CEAL HYDRAULIC TESTS PERFORMED
AT CHANCE VOUGHT AIRCRAFT, INCORPORATED
FOR JUNE, JULY, AUGUST 1960

S/N and Seal	Seal Measurement O.D.-(In.)	Seal Measurement Depth-(In.)	Cavity Measurement I.D.-(In.)	Cavity Measurement Depth-(In.)	Test Details
97 AM 350 .006 wall Electrolized	(Initial) .9890-.9900 (Post Test) .9918	 .0950-.0957 .0905-.0908	.9930 (17-4 plug and base) (Finish 12 rms) (Hardness-Rc 35)	.0868	Installed once and removed, electrolizing apparently does nothing to relieve initial scoring. Leakage very slight on impulse test. Stopped cycling at 7760 cycles to compare with S/N 98. Plug and base marked fairly deeply, but fretting corrosion absent or very slight. Installation torque is apparently reduced. Leakage rates: @3000 cycles, 1 cc/hr.; @5000 cycles, 1 cc/hr.; @7760 cycles, zero
	Squeeze: .0082 min.; .0089 max. Diametral Clearance: .0030 min.; .0040 max.				
98 AM 350 .006 wall Electrolized	(Initial) .9900-.9910 (Post Test) Ring Split	 .0940-.0944	.9930 (17-4 plug and base) (Finish 12 rms) (Hardness-Rc 43)	.0868	Installed without a back up ring and removed to see if back up affects initial scoring. Score marks were evident on removal. Leakage was slight throughout 7760 cycles. Seal was found to be split on removal. Plug and base had wear mark but no fretting corrosion. Installation torque is reduced. Leakage rates: @3000 cycles, 4.3 cc/hr.; @5000 cycles, 2.7 cc/hr.; @7760 cycles, zero.
	Squeeze: .0072 min.; .0076 max. Diametral Clearance: .0030 min.; .0040 max.				
99 AM 350 .010 wall Formed from Annealed Stock	(Initial) .9890-.9896 (Post Test)	 .0930-.0931	None		Compression tested - see App.II-7 for test results

SUMMARY OF HI-CEAL HYDRAULIC TESTS PERFORMED AT CHANCE VOUGHT AIRCRAFT, INCORPORATED FOR JUNE, JULY, AUGUST 1960

S/N and Seal	Seal Measurement O.D.-(In.)	Seal Measurement Depth-(IN.)	Cavity Measurement I.D.-(In.)	Cavity Measurement Depth-(In.)	Test Details
100 AM 350 .010 wall Formed from Annealed Stock	(Initial) .9905-.9908 (Post Test) .9938-.9940	 .0920-.0925 .0880-.0885	.9935 (17-4 plug, 416 base) (Finish 10 rms) (Hardness - Rc 43 plug, Rc 35 base)	.0860	Used a slip fit cavity to center seal on plug. Failed proof pressure at 450°F. Impulse cycled 119 times. Excessive leakage.
	Squeeze: .0060 min.; .0065 max. Diametral Clearance: .0027 min.; .0030 max.				
101 AM 350 .010 wall Formed from Annealed Stock	(Initial) .9892-.9897 (1st Post Test) .9927-.9968 (2nd Post Test) .9948-.9968	 .0925-.0935 .0872-.0875 .0855-.0865	(First Installation) 1.0040 (Second Installation) .9990 (17-4 plug, 416 base) (Finish 18 rms) (Hardness-Rc 43 plug, Rc 35 base)	 .0855 .0830	First installation - used a slip fit cavity to center seal on plug. Failed proof pressure at 450°F. No cycling done. Second installation - sustained proof pressure at 450°F. Impulse cycled 4417 times. Wear was very slight. Leakage rate at 4412 cycles, 68 cc/hr.

1st installation:
Squeeze: .0070 min.; .0080 max.
Diametral Clearance: .0143 min.; .0148 max.

2nd installation:
*Squeeze: .0095 min.; .0105 max.
*Diametral Clearance: .0093 min.; .0098 max.
*Computed from initial seal dimensions

SUMMARY OF HI-CEAL HYDRAULIC TESTS PERFORMED
AT CHANCE VOUGHT AIRCRAFT, INCORPORATED
FOR JUNE, JULY, AUGUST 1960

S/N and Seal	Seal Measurement O.D.-(In.)	Seal Measurement Depth-(In.)	Cavity Measurement I.D.-(In.)	Cavity Measurement Depth-(In.)	Test Details
102 AM 350 .010 wall As formed	(Initial) .9892-.9902 (1st Post Test) .9975-.9993 (2nd Post Test) .9997-1.005	 .0935-.0938 .0862 .0820-.0830	(First Installation) 1.0040 (Second Installation) 1.000 (17-4 plug, 416 base) (Finish 2-6 rms) (Hardness-Rc 43 plug, Rc 35 base)	 .0855 .0790	First installation - sustained proof pressure but leaked badly on impulsing. Total cycles was 2050. Second installation - very good sealing although not perfect. 17-4 plug was badly pitted, wear line was easily felt with finger nail. 416 base also had wear line which could be felt with finger nail, but corrosion pits were not as bad as on plug. Seal marked about the same as plug and base. Leakage rates: @2000 cycles. 2 cc/hr.; @5000 cycles, 2 cc/hr.; @8000 cycles, 10 cc/hr.; @11,000 cycles, 3.5 cc/hr.; @13,000 3.5 cc/hr.; @16,000 cycles, 7 cc/hr.; @21,750 cycles, 19 cc/hr.
	1st installation: Squeeze: .0080 min.; .0083 max. Diametral Clearance: .0138 min.; .0148 max. 2nd installation: *Squeeze: .0145 min.; .0148 max. *Diametral Clearance: .0098 min.; .0108 max.				
105 CRES 304 .015 wall As formed	(Initial) .9885-.9890 (1st Post Test) .9940-.9950 (2nd Post Test) 1.0030-1.0040	 .0942-.0950 .0907-.0918 .0833-.0838	(First Installation) 1.0000 (Second Installation) 1.0060 (17-4 plug and base) (Finish 15 rms) (Hardness-Rc 35)	 .0885 .0815	First installation - plug and base sanded with grit paper for rough finish. Leaked during 450°F proof pressure. No cycling. Second installation - leaked during 450°F proof pressure and cycling. Total cycles 2060.
	1st installation: Squeeze: .0057 min.; .0065 max. Diametral Clearance: .0108 min.; .0115 max. 2nd installation: *Squeeze: .0127 min.; .0135 max. *Diametral Clearance: .0168 min.; .0175 max.				

*Computed from initial seal dimensions

SUMMARY OF HI-CEAL HYDRAULIC TESTS PERFORMED
AT CHANCE VOUGHT AIRCRAFT, INCORPORATED
FOR JUNE, JULY, AUGUST 1960

S/N and Seal	Seal Measurement O.D.-(In.)	Seal Measurement Depth-(In.)	Cavity Measurement I.D.-(In.)	Cavity Measurement Depth-(In.)	Test Details
105B CRES 304 .015 wall Teflon coated	*(Initial) 1.0033-1.0046 (Post Test) 1.0051-1.0054	 .0838-.0840 .0763-.0781	1.0060 (17-4 plug, 416 base) (Finish 10-15 rms) (Hardness Rc 43 plug, Rc 35 base)	.0740	Zero leakage for 10,000 impulse cycles. Very slight leakage through 21,250 cycles. Wear on both 17-4 plug and 416 base could barely be felt with finger nail. There was no evidence of fretting corrosion on either surface. Teflon wore off of top, bottom, and back side of seal. Leakage rates: no leakage to 10,000 cycles; @12,000 cycles, 1.8 cc/hr.; @15,000 cycles, 1.8 cc/hr.; @18,000 cycles, 2.9 cc/hr.; @21,250 cycles, 2.7 cc/hr. Metallic residue was very slight because of reduced wear.
	Squeeze: .0098 min.; .010 max. Diametral Clearance: .0014 min.; .0027 max.				
113 CRES 304 .015 wall As formed	(Initial) 1.2391-1.2412 (Post Test) 1.2490-1.2515	 .0920-.0935 .0750-.0850	1.2532 (17-4 housing, 416 valve) (Finish 16-32 rms) (Hardness Rc 35)	.073 -.0808	Class B check valve and test housing. Unable to effect a seal at 450°F.
	Squeeze: .0112 min.; .0205 max. Diametral Clearance: .0120 min.; .0141 max.				

*This seal had been installed twice previously without a coating (ref S/N 105)

SUMMARY OF HI-CEAL TESTS FOR SEPTEMBER, OCTOBER, NOVEMBER 1960

S/M and Seal	Seal Measurement O.D.-(In.) - Depth-(In.)	Cavity Measurement I.D.-(In). Depth-(In.)	REMARKS
106 CRES 304 .012 Wall Teflon-Coated	(Initial) .9377-.9963 .0948-.0952 (Post Test) 1.0135-1.0137 .0869-.0890 Squeeze: .0103 min.; .0107 max. Diametral Clearance: .0174 min.; .0260 max.	1.0137 .0845 (17-4 plug, 416 base)* (Finish - 6-10RMS) (Hardness - C-37)	Total cycles - 13,945. Teflon coating was not bonded too well. Seal was worn irregularly. On one side the wear line was close to the inside lip. On the opposite side the wear line was in the middle of the seal. Failure occurred instantaneously. Seal was split along about 1/10 circumference. Leakage rates: @ 3000 cycles, 5.8 cc/hr.; @ 6000 cycles, 7.2 cc/hr.; @ 9000 cycles, 9.4 cc/hr.; @12,000 cycles, 6.3 cc/hr.; @ 13,945 cycles, 3.9 cc/hr.
115 CRES 304 .015 Wall as formed	(Initial) .9837-.9892 .0920-.0928 (Post Test) .0030-.0033 .0849-.0855 Squeeze: .0080 min.; .0083 max. Diametral Clearance: .0038 min.; .0043 max.	.0030 .0840 (440-C plug & base)* (Finish - 10-25 RMS) (Hardness - RC 37-40)	Sealed fairly good for 2,500 cycles. Failure of secondary seal at 12,150 cycles stopped test. Plug is worn but no fretting corrosion. Unable to feel wear ring with fingernail. Base plate is worn the same as the plug but there is more fretting corrosion. Seal surfaces are also pitted. Leakage rates: @ 3,000 cycles; 3.2 cc/hr. @ 6,000 cycles, 10.8 cc/hr.; @ 9,000 cycles, 12.1 cc/hr.; @ 11,000 cycles, 12.6 cc/hr.
116 CRES 305 .015 Wall as formed	(Initial) .9892-.9895 .0930-.0932 (Post Test) .9927-.9929 .0868-.0876 Squeeze: .0075 min.; .0077 max. Diametral Clearance: .0035 min.; .0038 max.	.9930 .0855 (416 Plug, 17-4 Base)** (Finish - 16-28 RMS) (Hardness - RC 34 Plug)	Installation sealed very good; total cycles - 35,411. Seal was in good shape after test and was not cracked or split. Secondary seal leak precluded leakage measurement. No smoke visible in cell which indicates slight secondary seal leakage, and even less Hi-ceal leakage. Plug wear ring is quite thin, with some pits visible only with microscope. Base is about the same as plug, even though it is 17-4.

* Test fixture utilizing base plate
** Test fixture without base plate

SUMMARY OF HI-CEAL TESTS FOR SEPTEMBER, OCTOBER, NOVEMBER 1960 - Continued

S/N and Seal	Seal Measurement O.D. (In.)	Seal Measurement Depth (In.)	Cavity Measurement I.D. (In.)	Cavity Measurement Depth (In.)	REMARKS
117 CRES 304 .015 Wall as formed	(Initial) .9890-.9895 (Post Test) .9994-1.000	(Initial) .0929-.0931 (Post Test) .0774-.0778	1.0010 (416 Plug, 17-4 Base)** (Finish - 7 RMS) (Hardness - RC 34)	.0761	Excellent sealing - applied 4,000 psi static pressure while cooling from 450°F to room temperature after 50,013 P.I.C. without leakage. The seal was sectioned and was found to be without evidence of cracking. Plug and base have wear ring with very slight putting which can be felt with fingernail. This is the first Hi-ceal which sustained 50,000 cycles. Leakage rates averaged 1.5 cc/hr. for 90% of cycling time. See for leakage curve.
	Squeeze: .0168 min.; .0170 max.				
	Diametral Clearance: .0115 min.; .0120 max.				
118 CRES 304 .015 Wall as formed	(Initial) .9882-.9885 (Post Test) .9928-.9930	(Initial) .0922-.0929 (Post Test) .0839-.0853	.9930 (17-4 Plug, Base)* (Finish - 6-10 RMS) (Hardness - RC 35-40)	.0820	Leaked at proof pressure leakage check at 450°F. P.I.C. 7,075 times. Leaked all the way. Plug had considerable fretting corrosion. Wear line could not be felt with fingernail. Base has some fretting corrosion but not as bad as plug. Leakage rates: @ 3000 cycles, 3.6 cc/hr.; @ 5000 cycles, 9.4 cc/hr.; @ 7000 cycles, 14.4 cc/hr.
	Squeeze: .0102 min.; .0109 max.				
	Diametral Clearance: .0045 min.; .0048 max.				
119 CRES 304 .015 Wall as formed	(Initial) .9890-.9892 (Post Test) .9915-.0025	(Initial) .0912-.0916 (Post Test) .0863-.0873	.9948 (17-4 Plug Electrolized)* (420 Base) (Finish - 10-22 RMS) (Hardness - RC 36)	.0820	Sealed not too good on P.I.C. and leaked badly during 450°F proof pressure leakage test. Electrolized plug had long and fairly deep scribe marks on the sealing surface. One of these marks fell under the Hi-ceal. In spite of this, the seal was pretty good. There was little evidence of fretting corrosion. Total cycles - 25,083. Leakage rates: @ 3000 cycles, 13.5 cc/hr.; @ 6000 cycles, 7.0 cc/hr.; @ 9000 cycles, 21.1 cc/hr.; @ 12,000 cycles, 7.9 cc/hr.; @ 15,000 cycles, 7.9 cc/hr.; @ 18,000 cycles, 7.6 cc/hr.; @ 21,000 cycles, 11.2 cc/hr.; @ 25,000 cycles, 12.6 cc/hr. presents the accumulated leakage versus time.
	Squeeze: .0092 min.; .0096 max.				
	Diametral Clearance: .0056 min.; .0058 max.				

* Test fixture utilizing base plate
** Test fixture without base plate

SUMMARY OF HI-CEAL TESTS FOR SEPTEMBER, OCTOBER, NOVEMBER 1960 - Continued

S/N and Seal	Seal Measurement O.D. (In.)	Seal Measurement Depth (In.)	Cavity Measurement I.D. (In.)	Cavity Measurement Depth (In.)	REMARKS
120 CRES 304 .015 Wall as formed	(Initial) .9889-.9891	.0928-.0931	.9908	.0732	Sealed nicely up to 12,000 cycles after slight leakage on 450°F proof pressure leakage test. Seal was split about half around the circumference on removal. Fretting corrosion on one side of the seal. The plug had considerable fretting corrosion pits but could not feel the wear line with fingernail. Base plate had fretting corrosion pits but not as many as plug. Leakage rates: @ 3000 cycles, 4.3 cc/hr.; @ 6000 cycles, 1.8 cc/hr.; @ 9000 cycles, 0.9 cc/hr.; @ 12,000 cycles, 1.6 cc/hr.; @ 15,000 cycles, 2.3 cc/hr.; @ 18,000 cycles, 10.8 cc/hr.; @ 21,000 cycles, 11.3 cc/hr.; @ 25,000 cycles, 13.7 cc/hr. Total cycles - 25,311. See for leakage curve.
	(Post Test) .9915-.9920 (Seal Split)	.0755-.0775	(17-4 Plug, 416 Base)* (Finish - 6-10 RMS) (Hardness - RC 35-43)		
	Squeeze: .0196 min.; .0199 max.				
	Diametral Clearance: .0017 min.; .0019 max.				
121 CRES 304 .017 Wall as formed	(Initial) .9873-.9891	.925 -.0930	NONE		Compression tested. See curve, Appendix II-7.
	(Post Test) .9992-1.0001	.747 -.0768			
123 CRES 304 .017 Wall as formed	(Initial) .9875-.9882	.0929-.0930			
	First Installation (Post Test) .9902	.0880-.0888	.9890 (440 Plug, 416 Base)*	.0853	Failed 450°F static pressure test.
	Squeeze: .0076 min.; .0077 max.				
	Diametral Clearance: .0008 min.; .0015 max.				
	Second Installation (Post Test) Not measured		.9948 (440 Plug, 416 Base)*	.0820	Failed 450°F static pressure test.
	Squeeze: .0109 min.; .0110 max.				
	Diametral Clearance: .0066 min.; .0073 max.				

* Test fixture utilizing base plate
Based on initial seal dimensions

SUMMARY OF HI-CEAL TESTS FOR SEPTEMBER, OCTOBER, NOVEMBER 1960 - continued

S/M and Seal	Seal Measurement O.D. (In.)	Seal Measurement Depth (In.)	Cavity Measurement I.D. (In.)	Cavity Measurement Depth (In.)	REMARKS
124 CRES 304 .017 Wall as formed	First Installation (Initial) .9892-.9894	.0928-.0930	.9980	.0855	Failed 450°F proof pressure.
	(Post Test) .9930-.9940	.0868-.0872	(440 Plug, 416 Base)* (Finish - 8 RMS) (Hardness - RC 35-37)		
	Squeeze: .0073 min.; .0075 max.				
	Diametral Clearance: .0086 min.; .0088 max.				
	Second Installation (Post Test)		.9948	.0820	Completed 5159 P.I.C. but leaked badly. Could feel wear ring with fingernail. Some fretting corrosion even though there were few cycles.
	Squeeze: .0108 min.; .0110 max.				
	Diametral Clearance: .0054 min.; .0056 max.				
127 CRES 304 .019 Wall as formed	(Initial) .9862-.9878	.0923-.0928	NONE		Compression tested - see curve, Appendix II-7.
	(Post Test) .9972-.9986	.0951-.0769			
128 CRES 304 .019 Wall Teflon-Coated	(Initial) .9810-.9818	.0917-.0922	.9905	.0815	Installation sealed very well through 34,000 cycles. Teflon was work off of sealing surface. Residue left in cavity had metallic particles indicating that teflon had been stripped several thousand cycles before removal. Seal was found to be cracked when sectioned. The plug had a wide wear ring with some fretting corrosion. Wear ring could be felt with fingernail. Base plate was in same condition as plug. Leakage rates: 0 to 13,150 cycles, 0.4 cc/hr.; 13,150 cycles to 21,700 cycles, 0.9 cc/hr.; 21,700 to 37,000 cycles, 2.5 cc/hr.; 37,000 to 44,395 cycles, 10.8 cc/hr. See for leakage curve.
	(Post Test) .9946	.0885-.0898	(17-4 Plug and Base)* (Hardness: Plug RC 43 Base RC 34) (Finish: 6-10 RMS)		
	Squeeze: .0102 min.; .0107 max.				
	Diametral Clearance: .0087 min.; .0095 max.				

* Test fixture utilizing base plate
Based on original seal dimensions

SUMMARY OF HI-CEAL TESTS FOR SEPTEMBER, OCTOBER, NOVEMBER 1960 - Continued

S/M and Seal	Seal Measurement O.D. (In.)	Seal Measurement Depth (In.)	Cavity Measurement I.D. (In.)	Cavity Measurement Depth (In.)	REMARKS
129 CRES 304 .019 Wall as formed	(Initial) .9880-.9885 (Post Test) .9905	.0924-.0926 .0845-.0873	.9905 (440 Plug, 416 Base)* (Finish: 6-10 RMS) (Hardness: RC 35-37)	.0815	Completed 15,931 P.I.C.;wear mark on plug and base was very large with considerable black radial scratches. Different from usual wear pattern. Wide movement apparently caused bad leakage. Static leakage at 4,000 psi and 450°F after 15,931 cycles was 7 cc/hr.
133 Titanium .014 Wall as formed	(Initial) .9905-.9920 (Post Test) .9948-.9952 Squeeze: .0075 min.; .0087 max. Diametral Clearance: .0060 min.; .0075 max.	.0930-.0942 .0888-.0891	.9980 (440 Plug, Base)* (Finish: 6-10 RMS) (Hardness: RC 37-40)	.0855	Seal was removed prematurely. Leakage was occurring during P.I.C. but not badly. Plug - no evidence of fretting corrosion. Wear line is thin and cannot be felt with fingernail. Base plate wear mark even less pronounced. One spot on base showed a somewhat irregular wear pattern.
135 Titanium .014 Wall Thin Anodize	(Initial) .9898-.9925 (Post Test) .9925-.9943 Squeeze: .0070 min.; .0075 max. Diametral Clearance: .0017 min.; .0044 max.	.0945-.0950 .090 -.092	.9942 (17-4 Plug, 416 Base)* (Finish: 10 RMS) (Hardness: Plug RC 43 Base RC 35)	.0875	Sealed very well up to 8,500 cycles then failed rapidly. Seal was out of round when installed. Anodize was worn off during impulsing. Wear on seal was fairly deep with some pitting. Plug and base both have wide wear ring. Seal apparently not centered before tightening. Some pitting but not too deep. Total cycles - 9,527. Leakage rates: 0 to 8500 cycles, 1.3 cc/hr.; 8500 to 9527 cycles, 70 cc/hr.
136 Titanium .014 Wall Thick Anodize	(Initial) .9905-.9910 (Post Test) .9930-.9935 Squeeze: .0077 min.; .0080 max. Diametral Clearance: .0032 min.; .0037 max.	.0952-.0955 .0899-.0935	.9942 (17-4 Plug, 416 Base)* (Finish: 6-8 RMS) (Hardness: Plug RC 43 Base RC 34)	.0875	Sealed fairly well thru 10,000 cycles. Considerable contamination from scrubbing off anodize and metal shavings. The seal was worn much deeper and wider than plug or base. Could not feel thin wear line which was on plug and base. No fretting corrosion on either plug or base. Leakage rates: 0-10,231, 2.6 cc/hr.; 10,231-12,779, 30.6 cc/hr.; total cycles - 12,779.

* Test fixture utilizing base plate

SUMMARY OF HI-CEAL TESTS FOR SEPTEMBER, OCTOBER, NOVEMBER 1960 - continued

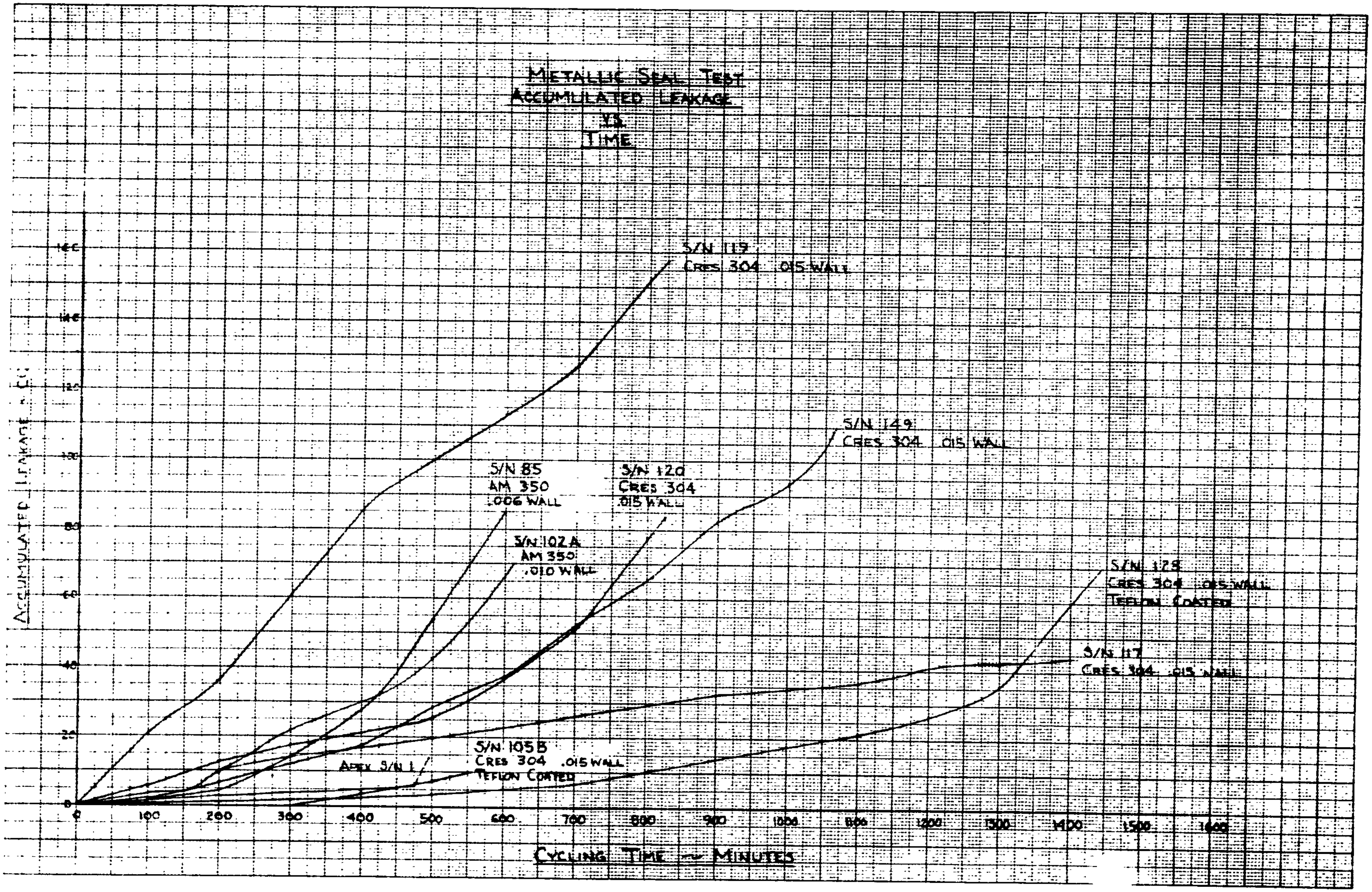
S/M and Seal	Seal Measurement O.D. (In.)	Seal Measurement Depth (In.)	Cavity Measurement I.D. (In.)	Cavity Measurement Depth (In.)	REMARKS
137 Titanium .014 Wall Moly Lubricant	(Initial) .9915-.9945 (Post Test) .9942-.9948	.0965-.0975 .0901-.0905	.9944 (420 Plug, 17-4 Base)* (Finish: 6-10 RMS (Hardness: Plug, RC 36 Base, RC 35)	.0889	Leaked continuously on impulse. Total cycles, 2,442. Molycote flaked off badly contributing to much contamination. Plug and base were both worn very deeply. Leakage rate was 34 cc/hr.
	Squeeze: .0076 min.; .0086 max. Diametral Clearance: .0000 min.; .0029 max.				
146 CRES 304 .015 Wall as formed	(Initial) .9899-.9905 (Post Test) .9975-.9930	.0938-.0940 .0858-.0860	1.0029 (416 Plug, 17-4 Base)** (Finish: 8 RMS) (Hardness: RC 34)	.0850	Completed 22,123 P.I.C. Considerable leakage toward end of cycles. Combination of large back as clearance and relatively low squeeze probably caused leakage. Leakage rate: 0-6600 cycles, 0 cc/hr.; 6600-10,960, 1.7 cc/hr.; 10,961-18,900, 3.5 cc/hr.; 18,900-22,123, 12.8 cc/hr. Static leakage at 4,000 psi and 450°F after 22,123 cycles was zero.
	Squeeze: .0088 min.; .0090 max. Diametral Clearance: .012[illegible] min.; .0130 max.				
147 CRES 304 .015 Wall as formed	(Initial) .9950-.9952 (Post Test) 1.0000-1.0050	.0940 .0778-.0779	1.0010 (416 Plug, 17-4 Base)** (Finish: 8 RMS) (Hardness: RC 34)	.0761	Completed 49,302 P.I.C. with negligible leakage. Zero leakage with 4,000 psi static pressure at 450°F after 49,302 cycles. Very good sealing with little wear. Seal was sectioned and no crack was found.
	Squeeze: .0179 Diametral Clearance: .0058 min.; .0060 max.				

* Test fixture utilizing base plate
** Test fixture without base plate

SUMMARY OF HI-CEAL TESTS FOR SEPTEMBER, OCTOBER, NOVEMBER 1960 - continued

S/M and Seal	Seal Measurement O.D. (In.)	Seal Measurement Depth (In.)	Cavity Measurement I.D. (In.)	Cavity Measurement Depth (In.)	REMARKS
149 CRES 304 .015 Wall as formed	(Initial) .9900-.9908 (Post Test) .9928-.9932 Squeeze: .0064 min.; .0065 max. Diametral Clearance: .0034 min.; .0042 max.	 .0939-.0940 .0892-.0900	.9942 (420 Plug, Base)* (Finish: 6-10 RMS) (Hardness: RC 36)	.0875	Sealing was fair at the start with only slight leakage at 4,000 psi and 450°F. The seal had more evidence of fretting corrosion than we have seen recently. Seal was found to be cracked when sectioned. Plug and base - pitting was obvious to naked eye. Wear line was thin but deep and could easily be felt with fingernail. Leakage rates: 0-3080 cycles, 1.3 cc/hr.; 3080-12,350, 3.1 cc/hr.; 12,350-20,800, 6.5 cc/hr.; 20,800-32,750, 9.5 cc/hr.; static leakage (after 32,750 P.I.C.) at 450°F and 4,000 psi was 1.6 cc/hr. presents an accumulated leakage curve for this seal test.
150 CRES 304 .015 Wall as formed	(Initial) .9900-.9902 (Post Test) .9943-.9950 Squeeze: .0084 min.; .0085 max. Diametral Clearance: .0078 min.; .0080 max.	 .0939-.0940 .0875-.0878	.9980 (420 Plug and Base)* (Finish: 8 RMS) (Hardness: RC 35)	.0855	Completed 22,123 P.I.C. Some fretting corrosion. No visible reason for high leakage. Leakage rate: 0-5000 cycles, 2.6 cc/hr.; 5000-18,400 cycles, 5.4 cc/hr.; 18,400-20,600, 9.0 cc/hr.; 20,600 - 22,123, 14 cc/hr. Static leakage at 4,000 psi and 450°F after 22,123 cycles 1.6 cc/hr.
151 CRES 304 .015 Wall as formed	(Initial) .9900-.9903 (Post Test) .9985-.9990 Squeeze: .0198 min.; .0200 max. Diametral Clearance: .0120 min.; .0123 max.	 .0938-.0940 .0805-.0810	1.0023 (17-4 Plug, 416 Base)* (Finish: (Hardness: RC 43, RC 34)	.0740	Completed 21,695 P.I.C. Some fretting corrosion wear could be felt with fingernail. Leakage rates: 0-4360 cycles, 0 cc/hr.; 4360-8700 cycles, 2.1 cc/hr.; 8700-13,200 cycles, 6.4 cc/hr.; 13,200-14,900 cycles, 20.8 cc/hr.;14,900-16,300 cycles, 4.8 cc/hr.; 16,300-21,695 cycles, 23.0 cc/hr. Static leakage at 4,000 psi and 450°F after 21,695 cycles was 5 cc/hr.

* Test fixture utilizing base plate



METALLIC SEAL TEST
ACCUMULATED LEAKAGE
VS
TIME
S/N 119
CRES 304 .015 WALL
S/N 149
CRES 304 .015 WALL
S/N 85
AM 350
.006 WALL
S/N 120
CRES 304
.015 WALL
S/N 102A
AM 350
.010 WALL
S/N 125
CRES 304 .015 WALL
TEFLON COATED
S/N 117
CRES 304 .015 WALL
S/N 105B
CRES 304 .015 WALL
TEFLON COATED
APEX S/N 1
ACCUMULATED LEAKAGE ~ CC
CYCLING TIME ~ MINUTES
0
20
40
60
80
100
120
140
160
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
1300
1400
1500
1600

APPENDIX III-1

GENERAL SPECIFICATION

FOR

THE PACKAGING OF HYDRAULIC COMPONENTS

MILITARY SPECIFICATION

HYDRAULIC COMPONENTS, MODULAR, GENERAL SPECIFICATION FOR THE PACKAGING OF

1. SCOPE

1.1 Scope - This specification covers the requirements that are common to packaging modular hydraulic components for use in Type III hydraulic systems conforming to MIL-H-8891.

1.2 Classification - Aircraft hydraulic systems in which packages covered by this specification are to be used shall be of the following type:

Type III: Pressure range: 0-4000 PSI

Temperature range: -65°F to +450°F

2. APPLICABLE DOCUMENTS

2.1 The following specifications, standards, drawings, and publications, of the issue in effect on date of invitation for bids, form a part of this specification to the extent specified herein:

SPECIFICATIONS

Federal

NN-P-515	Plywood, Container Grade
PPP-C-843	Cushioning Materials, Cellulosic
PPP-B-636	Boxes; Fiber, Solid, (for Domestic Shipment)
PPP-B-601	Boxes; Wood, Cleated - Plywood
PPP-B-621	Boxes; Wood, Nailed and Lock-Corner

Military

JAN-P-108	Packaging and Packing for Overseas Shipment-Boxes; Fiberboard, Exterior and Interior
MIL-P-116	Preservation, Methods of
MIL-B-131	Barrier Material, Water Vaporproof, Flexible

Military (cont)

MIL-C-5501	Closure, Aircraft, Tubing Protective
MIL-H-6083	Oil; Preservative, Hydraulic Equipment
MIL-I-6868	Inspection Process, Magnetic Particle
MIL-S-7742	Screw Threads, Standard Aeronautical
MIL-M-7911	Marking, Identification of Aeronautical Equipment, Assemblies, and Parts
MIL-H-8446	Hydraulic Fluid, Nonpetroleum Base, Aircraft
MIL-H-8891	Hydraulic Systems, Type III and Type IV, Installation, Design and Tests, Aircraft (General Specifications for)
MIL-D-70327	Drawings, Engineering and Associated Lists

Standards

MIL-STD-10	Surface Roughness, Waviness and Lay
MIL-STD-129	Marking for Shipment and Storage
MIL-STD-130	Identification Marking of U. S. Military Property
MIL-STD-143	Specification and Standards Use of
MS20995	Wire-Lock
MS33540	Safety Wiring - General Practices for
MS33568	Metals, Definition of Dissimilar
MS	
MS	
MS	MODULAR VALVES
MS	
MS	
MS	
MS	

Standards (Cont)

MS

MS

MODULAR VALVES

MS

MS

MS

MS

MS HI-SEALS

DRAWINGS

Air Force - Navy Aeronautical Standard Drawings

AND10050	Bosses, Standard Dimensions for Gasket Seal Straight Thread

(Copies of specifications, standards, drawings and publications required by contractors in connection with specific procurement functions should be obtained from the procuring activity or as directed by the contracting officer.)

3. REQUIREMENTS

3.1 General - In case of conflict between the requirements of this specification and a detail specification covering the requirements of a specific item of hydraulic equipment, the requirements of the detail specification shall take precedence over those of this specification. Additional requirements, applicable to a specific component, but not common to all components, will be covered by the detail specification.

3.2 Materials.- Materials shall conform to applicable specifications and shall be suitably processed. Materials which are not governed by applicable specifications may be used providing it can be demonstrated that their use will result in a superior product.

3.2.1 Metals - All metals used in the construction of manifolds to house modular hydraulic valves shall be compatible with the fluid, temperature, service, and storage conditions to which the component will be exposed. The metals shall possess corrosion-resistant characteristics on internal and external surfaces or shall be protected by the use of coatings to resist corrosion which may result from hydraulic fluid, dissimilar metal combinations, moisture, salt spray, and high-temperature deterioration as applicable.

3.2.1.1 Stabilization.- For structural and/or dimensional stability, a cold stabilization treatment is considered necessary operation in the fabrication of critical parts made from certain materials. Where such materials are used, the detail specification shall specify the materials and the processing required.

3.2.2 Plating

3.2.2.1 Chromium Plating - Chromium plating shall not be used on metallic seal seating surfaces.

3.2.3 Storage Life - All materials shall insure satisfactory service after normal storage.

3.2.4 Fabrication Practices and Standards - Specifications and standards for all materials, parts, processes, and equipment, which are not specifically designated herein and which are necessary for the

execution of this specification, shall be selected in the order of preference specified in Air Force - Navy Aeronautical Bulletin No. 143 except as provided in the following paragraph.

3.2.4.1 Standard Parts - Standard parts (MS, AN, or JAN) shall be used wherever they are suitable for the purpose, and shall be identified on the drawings by their part numbers. Commercial utility parts such as screws, bolts, nuts, cotter pins, etc., may be used, provided they possess required properties and are replaceable by the standard parts (MS, AN, or JAN) without alteration, and provided the corresponding standard part numbers are referenced in the parts list and, if practicable, on the contractor's drawing. In the event there is no corresponding standard part in effect on date of invitation for bids, commercial parts may be used provided they conform to all requirements of the detail specification.

3.3 Design and Construction

3.3.1 General - Manifolds designed to house modular hydraulic components may be individual parts, part of a hydraulic actuator, pump, or similar equipment, or part of the vehicle structure. The manifold may be designed to contain one or more modular components. Both the manifold and the package (the package consists of the manifold with its modular components installed) shall meet the test requirements noted herein. The following general design suggestions shall be considered.

(a) Determine which modular components are to be grouped into one manifold and prepare a schematic of the function this grouping is to perform.

(b) Determine allowable space envelope for the package.

(c) Determine what type accessibility to both component and package is required.

(d) Determine the direction hydraulic lines should enter and leave the manifold and select means of connecting hydraulic lines to the manifold.

(e) Determine means of mounting the package.

(f) Determine either before or during the design whether the manifold is to be a hog out, casting, or forging.

(g) The above information shall be utilized in arranging the components in relation to each other.

(h) Cost and weight savings shall be considered throughout the design.

3.3.2 Fluid.- The hydraulic fluid used shall be in accordance with MIL-H-8446.

3.3.3 Temperature Range - The manifold shall perform satisfactorily throughout the temperature range of -65°F to +450°F.

3.3.4 Metallic Seals - Metallic seals shall be used as the means of fluid pressure sealing. The type metallic seal used shall be in accordance with MS .

3.3.4.1 The manifold sealing surface shall have a minimum hardness of Rc34.

3.3.5 Cavity Dimensions - Dimensions and tolerances for standard modular cavities shall be in accordance with the applicable MS modular valve drawing of paragraph 2.1.

3.3.6 Ports - Type III manifold ports for connecting fittings, or hydraulic lines directly to the manifold, shall be equivalent, or superior, to the AND10050 Boss configuration used in Type I and II hydraulic systems. Where ports are not per AND10050, the configuration shall be subject to the approval of the procuring activity. Various types of port configurations which may be considered for Type III systems are itemized below:

(a) AND10050 boss in conjunction with acceptable packing suitable for the temperatures and pressures encountered.

(b) Modified AND10050 boss with redesigned sealing surface to accommodate the MS- metallic seal.

(c) Threaded integral male projection of the manifold in conjunction with suitable fittings.

(d) Special boss design to accommodate special fittings suitable to the specific application.

(e) Brazed joint utilizing brazed fittings to provide a permanent all brazed manifold-to-line joint.

(f) Face-mounted part utilizing the MS metallic seal for face-mounted manifolds.

Tube fittings and tubing, if incorporated into the manifold, shall conform to requirements in Specification MIL-H-8891.

3.3.6.3 Marking of Ports - All ports shall be clearly and permanently marked to indicate the proper connections to be made. Decalcomanias shall not be considered permanent marking.

3.3.6.4 Manifold Fluid Passages - Manifold fluid passages shall be routed in such a way as to require the minimum number of manifold ports. See Figure 1.

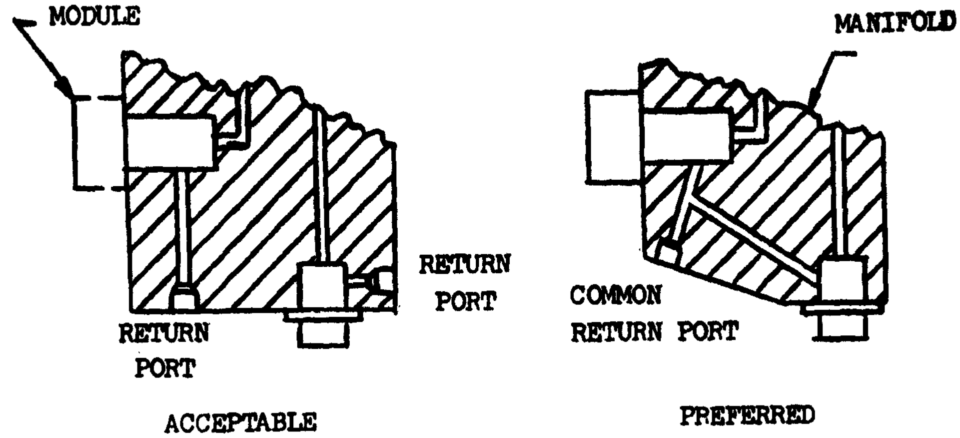


FIGURE 1

TYPICAL DESIGN

3.3.7 Bleeder Plugs - When required to meet the provisions of Specification MIL-H-8891, as applicable, for bleeding of entrapped air suitable bleeder plugs shall be provided at the highest practicable point in the manifold. Suggested types are AN814 plugs installed in bosses conforming to AND10050, AND10050 modified, or AND10067. Other types may be used subject to approval of the procuring activity.

3.3.8 Plugs - The use of permanent plugs in manifolds (sometimes called pin-plugs, or expansion type plugs) shall be held to a minimum consistent with good design. Caution shall be used in the manifold design when specifying location of expansion-type permanent plugs to avoid problems that may be caused by distortion and highly stressed areas within the manifold.

3.3.9 Threads - All threads shall be Class 3 threads in accordance with Specification MIL-S-7742.

3.3.10 Safetying - Provisions shall be made for safety wiring the components to the manifold. Safety wire shall be applied in accordance with the practice outlined in standard MS33540 and shall conform to drawing MS20995.

3.3.11 Surface Roughness - Surface roughness finishes shall be established and shall be specified on the manufacturer's assembly drawing as outlined in MIL-STD-10. All sealing surface machine finishes shall be held to the finishes specified by the applicable modular valve cavity MS drawing noted in paragraph 2.1.

3.3.12 Thread Lubricant - Any thread lubricant used to facilitate installation of the modular valves shall be compatible with the hydraulic fluid and shall not affect operation of components in the system.

3.3.13 Module Accessibility - Whenever practicable the manifold shall be designed so as to permit installation and removal of each module without requiring removal of adjacent modules. Adequate clearance for standard tools shall be provided. Accessibility of the complete package for installation and removal shall be a design requirement.

3.3.14 Module Attitude - The manifold shall be designed so as to position all MS modules in attitudes most suitable to the intent of the design. The one exception is that filters shall be mounted so the filter bowls are in the normally down position. See Figure 2.

3.3.15 Module Installation Torque - Installation torque requirements of many of the modular components are quite high; therefore, provisions for gripping the manifold when installing the valves shall be a design consideration.

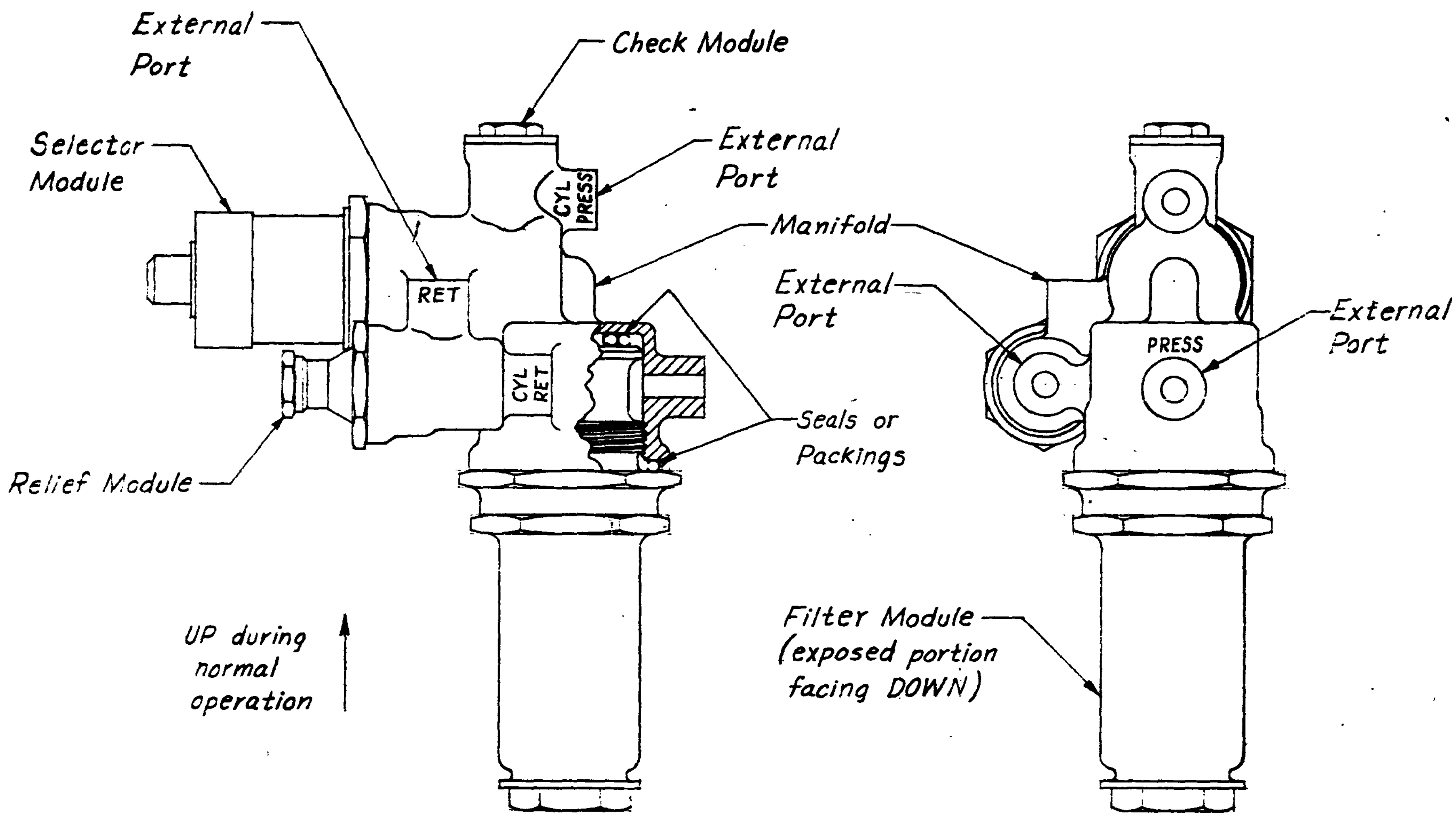


FIGURE 2 - Typical modular hydraulic package showing manifold, standard modular components and metallic seals or packings.

3.3.16 Structural strength - The manifold shall have sufficient strength to withstand all combinations of loads resulting from hydraulic pressure, temperature variations, and valve installation torques.

3.3.17 Weight - The weight shall be kept to a minimum consistent with good design, and shall be specified on the applicable drawing.

3.4 Special Tools - The manifold design shall not require the use of special tools for servicing the package.

3.5 Identification of Product

3.5.1 Marking - The information specified below shall be permanently marked on, or attached to, each manifold. The information marked in the spaces provided shall be in accordance with Specification MIL-M-7911 and in accordance with Standard MIL-STD-130.

MANIFOLD; MODULAR HYDRAULIC
Manufacturer's Part No.
Manufacturer's Serial No.

Decalcomanias are not considered permanent marking.

3.5.1.1 Each manifold shall be marked or tagged externally to indicate the standard replacement metallic seal and the quantities thereof used to affect sealing between the modular valves and the manifold.

3.6 Interchangeability - All parts having the same manufacturer's part number shall be directly and completely interchangeable with each other with respect to installation and performance. Changes in manufacturer's part numbers shall be governed by the drawing number requirements of Specification MIL-D-70327, or as otherwise specified by the procuring activity.

3.7 Workmanship

3.7.1 Quality - Workmanship shall be of sufficiently high quality to insure satisfactory operation and service life. The manufacturer shall exercise extreme care in fabricating, assembling, handling, and packaging valves to assure that the components are clean and free of contamination. All parts shall be free from pits, rust, scrapes, splits, cracks, burrs, and sharp edges.

3.7.2 Physical defect inspection - All magnetizable highly stressed parts shall be subjected to magnetic inspection per Specification MIL-I-6868. Cracks or other injurious defects disclosed by the inspection shall be cause for rejection. All non-magnetizable highly stressed parts shall be subjected to fluorescent penetrant inspection per Specification MIL-I-6866. Cracks or other injurious defects disclosed by the inspection shall be cause for rejection.

3.8 Performance

3.8.1 The manifolds shall satisfy the tests specified in paragraph 4.2.2.1.

3.8.2 The package (manifold plus modular components) shall satisfy the tests specified in paragraph 4.2.2.2.

4. QUALITY ASSURANCE PROVISIONS

4.1 Classification of Tests - The inspection and testing shall be classified as follows:

(a) Preproduction tests: Preproduction tests are those tests accomplished on samples, as specified in 4.2.1, to determine that the design of the production item meets the requirements of this specification.

4.1 Classification of Tests (cont)

(b) Acceptance tests: Acceptance tests are those tests performed on individual lots which have been submitted for acceptance.

4.2 Preproduction tests - Preproduction tests shall be performed on both the manifold and the package as designated herein.

4.2.1 Sampling Instruction - Preproduction tests on the manifold and the package shall be conducted using a representative sample of the production unit.

4.2.2 Tests

4.2.2.1 Manifold - The preproduction tests of manifolds shall consist of the following tests, as described under "Test methods".

(a) Examination of product per 4.5.1.1

(b) Proof pressure per 4.5.1.2

(c) Endurance in accordance with 4.5.1.3

(d) Burst Pressure in accordance with 4.5.1.3

4.2.2.2 Package - The preproduction tests of the package shall consist of the following tests as described under "Test Methods".

(a) Examination of product per 4.5.2.1

(b) Proof pressure and leakage per 4.5.2.2.1

(c) External and internal leakage per 4.5.2.3.1 and 4.5.2.3.2.1

(d) Pressure drop per 4.5.2.4

(e) Extreme temperature package functioning tests per 4.5.2.5.1, 4.5.2.5.2, 4.5.2.5.3

(f) Endurance per 4.5.2.6

4.2.3 Rejection and Retest of Preproduction Samples - Failure of any manifold subjected to the preproduction tests shall be cause for rejection of the manifold design represented.

4.3 Acceptance tests

4.3.1 Sampling - Each manifold and package submitted for acceptance under contract shall be subjected to the acceptance tests specified herein.

4.3.2 Tests

4.3.2.1 Manifolds - Acceptance tests of manifolds shall consist of the following tests, as described under "Test Methods":

(a) Examination of product per 4.5.1.1

(b) Proof pressure and leakage per 4.5.1.2

4.3.2.2 Packages - Acceptance tests of the packages shall consist of the following tests, as described under "Test Methods":

(a) Examination of product per 4.5.2.1

(b) Proof Pressure per 4.5.2.2.1

(c) External and internal leakage per 4.5.2.3.1 and 4.5.2.3.2.2

(d) Function test per 4.5.2.5.4

4.3.3 Rejection and retest - Failure of any manifold or package to conform to any of their acceptance tests shall be cause for rejection of that manifold or package and the lot represented. Manifolds or packages which have been rejected may be reworked to correct the defects found in the original and resubmitted for acceptance. Before resubmitting, full particulars concerning previous rejection and the action taken to correct the defects found in the original shall be furnished the Inspector. Units rejected after retest shall not be resubmitted without the specific approval of the procuring activity.

4.4 Test Conditions

4.4.1 Optional procedures - At the option of the procuring activity, any of the test requirements specified herein may be modified or waived owing to design or operating considerations. Request for modification or waiver of test requirements must be accompanied by complete detailed information.

4.4.2 Test fluid - Unless otherwise specified by the procuring activity, all preproduction and acceptance tests shall be performed with oil conforming to Specification MIL-H-8446.

4.4.3 Temperatures - Except where otherwise specified, the tests of this specification shall be conducted at a room temperature of 50° to 110°F. Oil temperature may range from 70° to 130°F. In preproduction tests the approximate temperature, except where specific temperatures are required, during the tests shall be recorded in the report.

4.5 Test Methods - The following tests are separated into those applying to the manifold only and to those applying to the package only.

4.5.1 Manifold

4.5.1.1 Examination of product - Each manifold shall be carefully examined to determine conformance with the requirements of this specification for weight, workmanship, marking, conformance of dimensions to applicable drawings, and for any visible defects. The determination of surface finish shall be made with a profilometer, comparator brush analyzer, or equally suitable comparison equipment with an accuracy of ±5 micro-inches at the level being measured.

4.5.1.2 Proof Pressure - The manifold shall have suitable plugs at all modular component cavities and porting other than the pressure port. A hydraulic proof pressure equal to 1.5 times the normal system operating pressure shall be applied at least two successive times and held for two minutes for each application. There shall be no evidence of external leakage (other than a slight wetting insufficiant to form a drop), excessive distortion, or permanent set. This test shall be performed at 95 ± 15°F for acceptance tests and 450°F for qualification tests.

4.5.1.3 Burst Pressure - The manifold shall withstand a burst pressure of 2.5 times the normal system operating pressure, and shall be applied at a maximum pressure rise rate of 25,000 psi per minute. The manifold shall not rupture under this pressure. This shall be the last test performed because of its destructive nature.

4.5.1.4 Endurance - The manifold shall be subjected to a hydraulic impulse test in accordance with the requirements of the detail specification to establish reliability of the fatigue strength. The detail specification shall indicate number of impulse cycles, schedule of cycling, cycle rate, temperature and impulse peaks. The number of cycles selected shall be applicable to the duty cycle over the anticipated life of the particular aircraft.

4.5.1.5 External Leakage - During the course of testing the manifold, external leakage other than a slight wetting insufficient to form a drop, shall be cause for rejection.

4.5.2 Package (Manifold with modular components installed)

4.5.2.1 Examination of Product - Each package shall be carefully examined to determine conformance with the requirements of this specification for design, workmanship, markings, applicable drawings and for any visible defects. Particular examination shall be made to insure that the modular component flanges are bottomed out on the manifold. The manufacturer's drawings and the manufacturer's applicable specification which were approved when qualification or approval were granted will be used by the Inspector as necessary, to determine that the packages submitted for acceptance under contract are identical to the design approved by the procuring activity.

4.5.2.2 Pressure Tests

4.5.2.2.1 Proof Pressure - A proof pressure equal to 1.5 times the normal system pressure shall be applied at least two successive times and held 2 minutes for each application. The components shall be operated through their normal function between applications of the test pressure. There shall be no external leakage, other than a slight wetting insufficient to form a drop, excessive distortion, or permanent set. This test shall be performed at $95 \pm 15^{o}F$ for acceptance tests and $450 \pm 15^{o}F$ for qualification tests.

4.5.2.3 Leakage Tests

4.5.2.3.1 External Leakage - There is no specific test for external leakage, but during the course of all tests listed in this specification, external leakage other than a slight wetting, insufficient to form a drop, shall be cause for rejection.

4.5.2.3.2 Internal Leakage

4.5.2.3.2.1 Preproduction Tests - These tests shall be performed with the package held in the position most conducive to leakage. The package shall be tested for internal leakage by applying 5 psi, 50% of working pressure, and working pressure for a period of 30 minutes each, unless otherwise specified in the detail specification. The leakage measurement period shall begin 2 minutes after the application of the required pressure. The package components shall be actuated between pressure applications. The rates of internal leakage shall not exceed those specified in the detail specification.

4.5.2.3.2.2 Acceptance Tests - These internal leakage tests shall be performed with the package held in the position most conducive to leakage. Pressures of 5 psi and working pressure shall be held for a period of 5 minutes each, unless otherwise specified in the detail specification. In each case the leakage measurement shall consist of the last 3 minutes of the 5-minute period. The rate of leakage shall not exceed that specified in the detail specification for the Preproduction test. Low rate, not readily measureable seepage, shall not be considered as leakage.

4.5.2.4 Pressure Drop - Pressure-drop characteristics of the package for a flow range of 0 to 150 percent of rated flow as prescribed in the detail specification shall be determined for the package. A piezometer or manometer across the package may be used for accurate measurement where the pressure-drop range is low enough to permit its use. The pressure drop observed at rated flow shall not exceed the value permitted by the applicable detail specification.

4.5.2.5 Extreme Temperature Package Functioning Tests

4.5.2.5.1 Low Temperature - The package shall be connected to a static head of 1 to 3 feet of the test fluid, or rated working pressure, whichever is the more critical condition. This arrangement shall be maintained at a temperature not warmer than -65°F for 3 hours after the temperature has stabilized at -65°F. After this period, the package shall be operated at least two times. Variation of intended function shall not exceed that permitted by the detail specification. The Acceptance tests prescribed for leakage shall be performed after each operation, and the requirements of the detail specification satisfied.

4.5.2.5.2 Intermediate Temperatures - Immediately following the Low temperature test (4.5.2.5.1), the test arrangement shall be warmed rapidly to a temperature to a temperature of 450°F. While the temperature is being raised, the package shall be operated its function at maximum increments of 75°F to determine satisfactory operation throughout the temperature range. These check tests shall be made without waiting for temperature of the entire package to stabilize.

4.5.2.5.3 High Temperature - The temperature shall be maintained at 450°F for a length of time sufficient to allow all parts of the package to attain the temperature. In no case shall the temperature at which this test is conducted be less than 430°F. The package shall then be operated its function at least two times. Variation of intended function from room temperature function shall not exceed that permitted by the detail specification. The Acceptance

test for Leakage (4.5.2.3.2.2) shall be performed after each operation, and the requirements of the detail specification satisfied.

4.5.2.5.4 Acceptance Test Package Functioning - The package shall be operated at least two times at room temperature as described in the detail specification. The Acceptance test for leakage (4.5.2.3.2.2) shall be performed after each operation, and the requirements of the detail specification satisfied.

4.5.2.6 Endurance - The package shall be subjected to cyclic operation and, where applicable, other fatigue tests such as hydraulic impulse in accordance with the requirements of the detail specification, which shall indicate number of cycles, schedule of cycling, cycle rate, stroke, rate of flow, loads, temperature, impulse peaks, etc. The number of cycles selected shall be applicable to the duty cycle over the anticipated life of the particular aircraft, but shall be not less than the values specified in table II, which is presented as a general guide. When applicable, leakage shall be tested at 25, 50, 75, and 100 percent of the number of cycles required. The general endurance test cycle of the package shall be governed by the requirements listed in paragraph (4.5.2.6.1). At the conclusion of the endurance test, the package shall operate satisfactorily and shall be disassembled and carefully inspected. There shall be no evidence of excessive wear in any part of the components or manifold.

TABLE II

Endurance Test

Type and Usage of Package	Cycles of Endurance Test
Emergency	5,000
Infrequent (less than 10 cycles per flight)	20,000
Frequent (more than 10 cycles per flight)	50,000
Flight control, steering, antiskid, etc.	See detail specification

4.5.2.6.1 General Endurance Tests Cycle - The endurance test that is specified shall be cycled over the following temperature spectrum.

(a) Rate of temperature rise or decay may vary within the shaded areas shown in Figure 3.

(b) Six and one half hours of endurance cycling are to be run in one day. The package shall be soaked overnight at the low temperature required to start the following day of testing.

(c) The first, middle and last repetitions of this spectrum begin with fluid temperature of -65°F. All other runs shall begin with the fluid temperature at ambient (70° to 100°F).

(d) The ambient temperature shall be maintained between 450 - 650°F during the time from the 2nd hour through 5-1/2 hours of the spectrum shown.

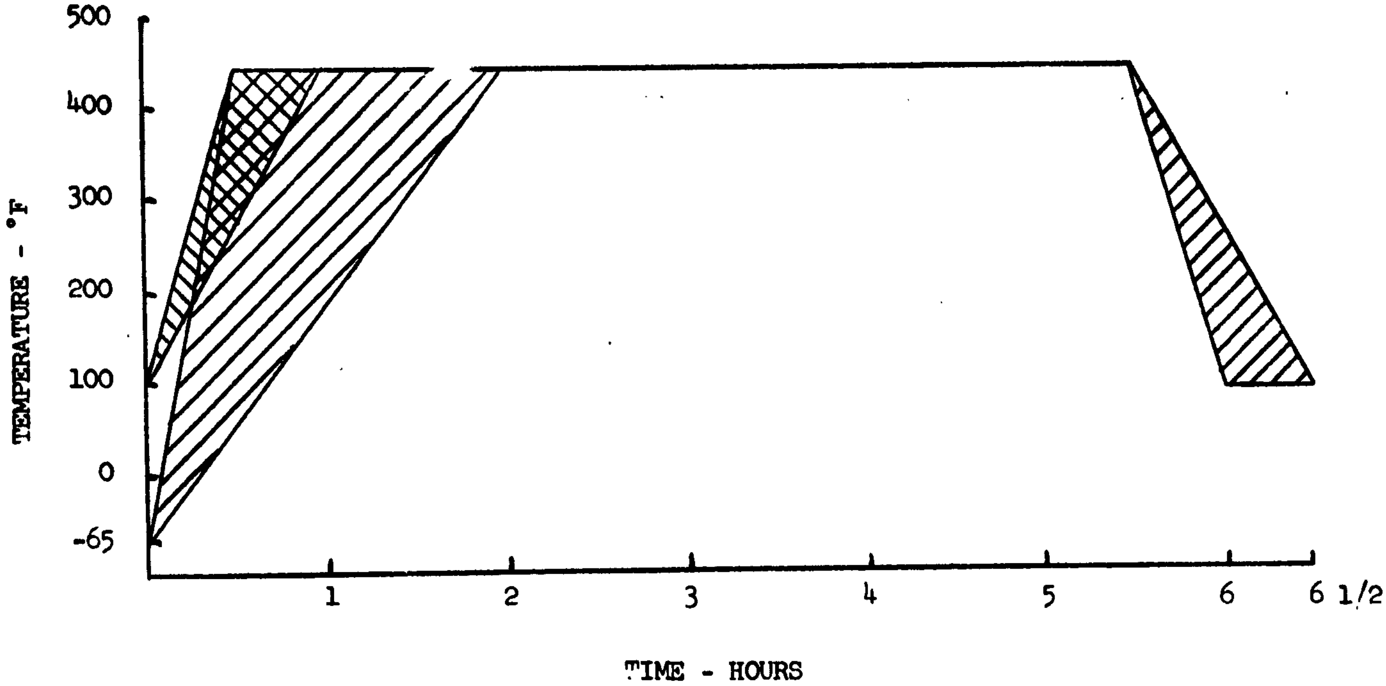


Figure 3
Time - Temperature Spectrum

5. PREPARATION FOR DELIVERY

5.1 Application - The requirements of Section 5 apply only to direct purchases by or direct shipment to the Government.

5.2 Preservation and packaging - Unless otherwise specified, each manifold shall be protected from corrosion and packaged in accordance with Specification MIL-P-116, Method 1A-8.

5.2.1 All cavities of the manifold shall be flushed with corrosion-preventive oil conforming to Specification MIL-H-6083 and the excess oil drained, taking care that enough oil remains in the cavities to completely cover all surfaces. All ports shall be sealed with closures conforming to Specification MIL-C-5501, or other acceptable seals.

5.2.2 Each manifold shall be cushioned with cellulose wadding conforming to Specification PPP-C-843, type III, sealed in a bag constructed of material conforming to Specification MIL-B-131 and overpackaged in a snug-fitting fiberboard container.

5.3 Packing

5.3.1 Immediate use shipment - Containers shall conform to that quality standard commercially utilized by the industry.

5.3.2 Domestic shipment - Domestic exterior shipping containers shall conform to Specification PPP-B-636, or PPP-B-601. When fiberboard exterior shipping containers are used, such containers shall be fabricated from fiberboard having a Mullen test of 275 pounds or more.

5.3.3 Overseas shipment - Overseas exterior shipping containers shall conform to Specification PPP-B-601, PPP-B-621, or JAN-P-108. When plywood is used, unless otherwise specified, it shall conform to Specification NN-P-515, type II, class 2.

5.4 Marking of shipments - Interior packages and exterior shipping containers shall be marked in accordance with Standard MIL-STD-129. The identification shall be composed of the following information listed in the order shown:

Stock No. or other identification number as specified in the purchase document.*

MANIFOLDS; AIRCRAFT, MODULAR HYDRAULIC

Specification MIL-H-

Quantity

Date of manufacture or date of preservation

Manufacturer s Part No.

Manufacturer s name or trade-mark

Name of contractor (if different from manufacturer)

*NOTE: The contractor shall enter the Federal Stock No. specified in the purchase document or as furnished by the procuring activity. When the Federal Stock No. is not provided or available from the procuring activity, leave space therefor and enter the Stock No. or other identification as provided by the procuring activity.

6. NOTES

6.1 Intended Use - Manifolds covered by this specification are intended for use in aircraft modular hydraulic systems, conforming to Specification MIL-H-8891. The package or manifold should not be used with any hydraulic fluid other than that conforming to Specification MIL-H-8446, as applicable, unless otherwise specified by the procuring activity.

6.2 Definitions - For the purpose of this specification, the items below are defined as follows:

(a) "Modular Component" - a standard MS modular hydraulic flow control, pressure indicator, fluid cleaning, relief or control valve.

(b) "Manifold" - a housing which contains and interconnects two or more MS modules.

(c) "Package" - a complete assembly consisting of one or more manifolds with MS modular components and seals installed in the manifold (s).

6.3 Qualification - With respect to products requiring qualification, awards will be made only for such products as have, prior to the time set for opening of bids, been tested and approved for inclusion in the applicable Qualified Products List whether or not such products have actually been so listed by that date. The attention of the suppliers is called to this requirement, and the manufacturers are urged to arrange to have the products that they propose to offer to the Federal Government, tested for qualification, in order that they may be eligible to be awarded contracts or orders for the products covered by this specification. The activity responsible for the Qualified Products List is the Bureau of Naval Weapons, Navy Department, Washington 25, D. C.; however, information pertaining to qualification of products may be obtained from the Commanding Officer, U.S. Naval Air Material Center, Naval Base, Philadelphia 12, Pennsylvania.

NOTICE: When Government drawings, specifications, or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the Government may have formulated, furnished, or any way supplied the said drawings, specifications, or other data is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or

corporation, or conveying any rights or permission to manufacture, use, or sell any patented invention that may in any way be related thereto.

Custodians:

Navy - Bureau of Naval Weapons

Air Force

Preparing Activity:

Navy - Bureau of Naval Weapons

APPENDIX III-2

TEST PROCEDURES FOR MODULAR HYDRAULIC PACKAGES

AER-AVO-53720-181, Package #1
AER-AVO-53720-182, Package #2
AER-AVO-53720-183, Package #4
AER-AVO-53720-184, Package #5

CHANCE VOUGHT AIRCRAFT
INCORPORATED · DALLAS, TEXAS

ORIGIN. BY A. R. Wilson	ORIGIN. GP. APP.	PROJECT ENGINEER		CVA AER-AVO-53720 -0-181
GROUP UNIT 53725	DATE	DATE	DATE	PAGE 1 OF 7
REL. DATE ____ REL. DESK ____	DATE	DATE	DATE	ECP PC

TEST PROCEDURE

FOR
PACKAGE, MODULAR HYDRAULIC #1

3 MODULE CONTROL

CVS-54175-1

#1 PACKAGE
TEST PROCEDURE

1. SCOPE

1.1 Scope - The test procedure covers the testing requirements for the #1 modular hydraulic package. This package consists of the CVS-54175-1 Manifold and three modular valves.

1.2 Classification - The #1 package is classified as a Class A package, indicating a flow range from 0 to 4 GPM.

2. APPLICABLE DRAWINGS - The following drawings of the issue in effect on date of this test procedure form a part of the test procedure:

2.1 Drawings -

CVS-54175	CVP-3345
CVP-3344	CVP-3355

3. TEST PROVISIONS

3.1 Test Fluid - The test fluid shall be MLO-8200

3.2 Test Fluid Temperature - Tolerance on fluid temperatures specified herein shall be ±5°F. The manifold outlet fluid temperature shall be specified, and the manifold inlet fluid temperature may be decreased a maximum of 25°F to compensate for heat generation. The fluid temperatures shall be measured as near as practicable to the outlet ports.

3.3 Filtration - The test fluid shall be continuously filtered through a filter capable of removing all particles in excess of 25 microns (25microns absolute), and shall remove 98% of all particles larger than 10 microns. The filter and element used shall be satisfactory for the temperature range encountered, and cleaned or changed regularly to prevent clogging. (Element shall conform to MIL-F-8815 testing procedures.)

3.4 Contamination - No contaminating agents shall be purposely added to the test fluid.

3.5 Test Package - The test package shall be partly or fully assembled, as dictated by the applicable paragraph, with provisions in the test set-up to cause each module to operate throughout its full cycle. All modules shall be checked for proper installation torque and safety wiring. The test package shall be set up per Figure 1 or Figure 2, as applicable.

4. TEST PROCEDURE

4.1 Module Function and Leakage Tests

4.1.1 Set up test apparatus per Figure 1, leaving the external port to the

CVP-3355 selector valve open to atmosphere. With CVP-3355 selector valve de-energized, pressure selected to line #1 and test fluid at +450°F, apply hydraulic pressure by hand pump until pressure gage G_1 reads 4,000 psi $\pm$20 psi. At this pressure there shall be no appreciable leakage at the manual selector vent line or at the CVP-3355 selector valve inlet port. Reduce pressure to zero. Reposition the manual selector valve to apply hydraulic pressure to line #2 and apply sufficint hydraulic pressure (approximately 2 to 8 psi) by hand pump to cause the CVP-3345-1 check valve poppet to unseat (evident by flow through vent to a container) Repeat this cycle 24 times.

4.1.2 With test set-up same as in paragraph 4.1.1 and test fluid at room temperature, select hydraulic pressure to line #1 and maintain 5 psi hydraulic pressure for five minutes. The leakage rate past the check valve during the last three minutes shall not exceed 3 drops per minute. Increase hydraulic pressure to 4,000 psi and maintain this pressure for five minutes. The leakage rate past the check valve during the last three minutes shall not exceed 1 drop per minute.

4.1.3 Repeat paragraph 4.1.1 with test fluid at room temperature.

4.1.4 Repeat paragraph 4.1.1 with test fluid at -20°F.

4.1.5 Set up test apparatus as shown in Figure 2 and select hydraulic pressure to line #1 and by-pass the cylinder assembly. Make limit switches inoperative and energize CVP-3355 selector valve. With test fluid at -20°F, and system bled of air, apply hydraulic pressure until flow-meter indicates a 6 gpm flow. At this flow rate, the differential pressure between gages G_1 and G_2 should be approximately 2,100 psi. Record the actual differential pressure. De-energize CVP-3355 selector valve and select hydraulic pressure to line #2. Apply hydraulic pressure until flow meter indicates a 12 gpm flow. At this flow rate, the differential pressure between G_2 and G_3 should be approximately 2,150 psi. Record the actual differential pressure. Repeat paragraph 4.1.5 cycle 24 times. There shall be no appreciable leakage externally around modules.

4.1.6 Repeat paragraph 4.1.5 with test fluid at room temperature.

4.1.7 Repeat paragraph 4.1.5 with test fluid at +450°F.

4.1.8 Alter test set-up of Figure 2 by leaving the external port to the CVP-3355 selector valve open to atmosphere. Select hydraulic pressure to line #2, close shut-off valve #1, and de-energize CVP-3355 Selector Valve. With test fluid at room temperature apply 400 psi hydraulic pressure and maintain this pressure for three minutes. The leakage rate past the selector valve port shall not exceed 40 drops per minute.

4.1.9 Alter test set-up of Figure 2 by closing shut-off valve #2 and removing the CVP-3345 Check Valve, leaving the check valve cavity open to atmosphere. With the CVP-3355 Selector Valve energized and test fluid at room temperature, select hydraulic pressure to line #1. Apply 4,000 psi hydraulic pressure and maintain this pressure for three minutes. The leakage rate at the check valve cavity shall not exceed 80 drops per minute.

4.2 Proof Pressure Tests

4.2.1 With test apparatus set up as shown in Figure 2 (all modules installed), close shut-off valves #1 and #2 and energize CVP-3355 Selector Valve. With test fluid at room temperature and pressure selected to line #1, apply hydraulic pressure at a rate not exceeding 25,000 psi per minute until pressure gage G_2 reads 6,000 psi ± 20 psi. Maintain this pressure for two minutes. There shall be no evidence of excessive leakage, permanent set, or other damage to the package.

4.2.2 Replace the CVP-3355 Selector Valve with the selector valve test cap. Maintain shut-off valves closed and pressure selected to line #1. Apply hydraulic pressure until pressure gage G_2 and G_3 read 6,000 psi ± 20 psi. Maintain this pressure for two minutes. There shall be no evidence of excessive leakage, permanent set, or other damage to the package.

4.3 Cycling Tests

4.3.1 Set up test apparatus as shown in Figure 2 with limit switches set to permit the maximum permissable piston travel in extension and retraction without permitting piston bottoming in either extreme position. Limit switches shall simultaneously operate the CVP-3355 selector valve in the No. 1 package and the four way-two position solenoid operated selector valve in the test set up. Wire these selector valves so that when line #1 is pressurized the CVP-3355 selector valve shall be energized. With test fluid at room temperature and the manual four way-two position selector valve positioned to operate the hydraulic cylinder, supply 4,000 psi ± 20 psi hydraulic pressure at a flow rate of approximately 4 gpm. Operate at this pressure and flow rate for a minimum of 100 cycles. At the completion of at least 100 cycles, maintain pressure and increase the flow rate to 8 gpm. Operate at this pressure and flow for a minimum of 100 cycles. There shall be no evidence of excessive leakage or module mal-functioning.

4.3.2 Repeat paragraph 4.3.1 with test fluid at +450°F.

4.4 Pressure Drop Tests

4.4.1 Alter test set-up of Figure 2 by setting manual selector to cause cylinder to be by-passed. Select hydraulic pressure to line #2 and de-energize CVP-3355 selector valves. With test fluid at room temperature apply hydraulic pressure to obtain 1,2,3 and 4 gpm flow at the flowmeter. Record pressure readings at G_2 and G_3 for these flow values.

4.4.2 Select hydraulic pressure to line #1 and energize CVP-3355 Selector Valve. With test fluid at room temperature apply hydraulic pressure to obtain 1,2,3 and 4 gpm flow at the flowmeter. Record pressure readings at G_1 and G_2 for these flow values.

4.4.3 Alter test set-up of Figure 2 by removing the three modular valves, replacing them with the respective test plugs. With shut-off valve #1 closed and shut-off valve #2 opened maintain hydraulic pressure to line #1. With test fluid at room temperature apply hydraulic pressure to obtain 1,2,3 and 4 gpm flow at the

flowmeter. Record pressure readings at G_1 and G_2 for these flow values.

4.4.4 Alter test set-up of paragraph 4.4.3 by capping the return port of the 4W-2P Selector Valve and opening shut-off valve #1. With test fluid at room temperature, select hydraulic pressure to line #2 and apply hydraulic pressure to obtain 1,2,3 and 4 gpm flow at the flowmeter. Record pressure readings at G_2 and G_3 for these flow values.

FIGURE 1

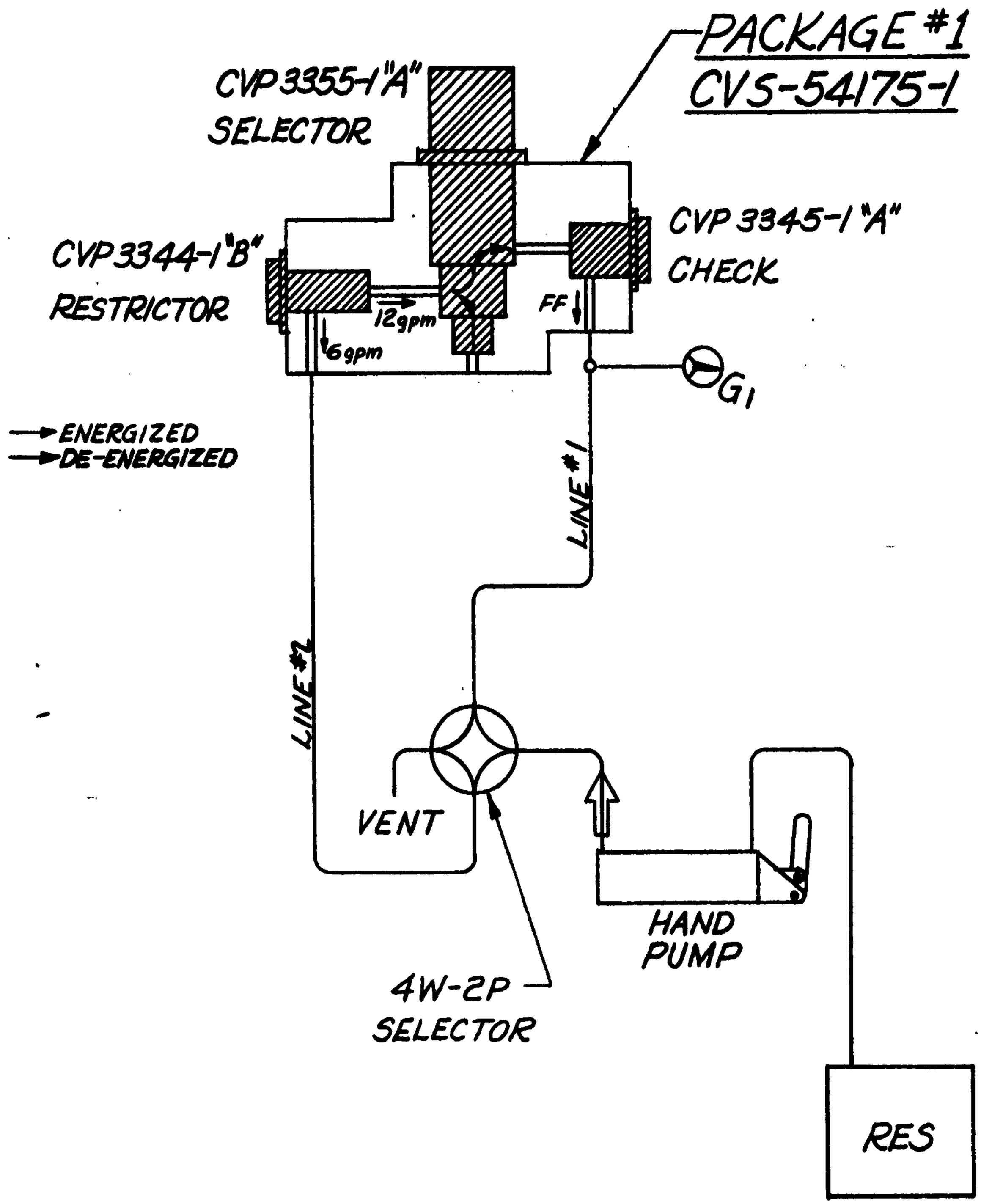
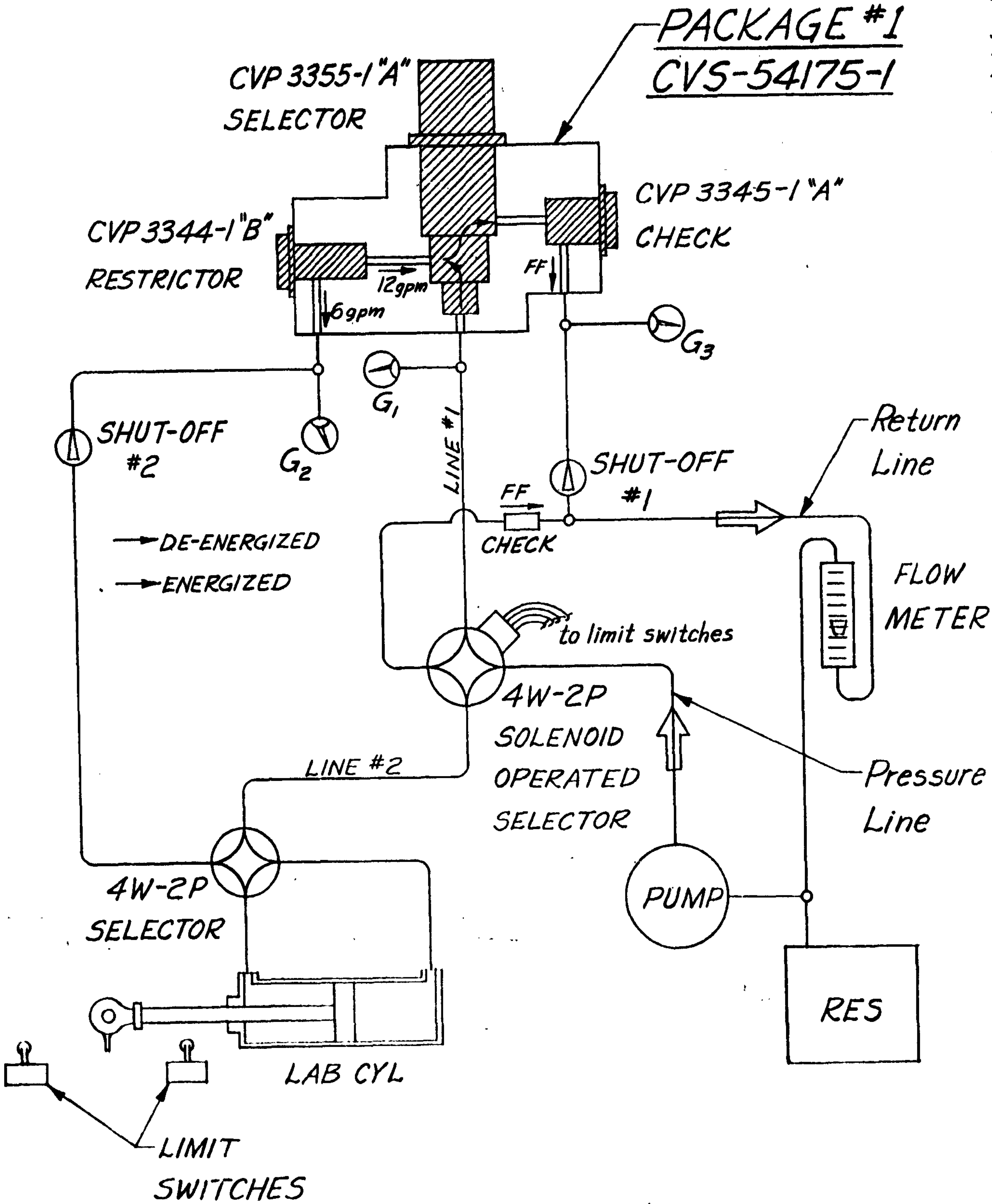

PACKAGE #1
CVS-54175-1
CVP 3355-1 "A"
SELECTOR
CVP 3345-1 "A"
CHECK
CVP 3344-1 "B"
RESTRICTOR
12gpm
6gpm
FF
G1
ENERGIZED
DE-ENERGIZED
LINE #1
LINE #2
VENT
HAND
PUMP
4W-2P
SELECTOR
RES

FIGURE 2


PACKAGE #1
CVS-54175-1
CVP 3355-1 "A"
SELECTOR
CVP 3345-1 "A"
CHECK
CVP 3344-1 "B"
RESTRICTOR
12gpm
6gpm
FF
G1
G2
G3
SHUT-OFF
#2
LINE #1
SHUT-OFF
#1
FF
CHECK
Return
Line
FLOW
METER
-DE-ENERGIZED
-ENERGIZED
to limit switches
4W-2P
SOLENOID
OPERATED
SELECTOR
Pressure
Line
LINE #2
4W-2P
SELECTOR
PUMP
RES
LAB CYL
LIMIT
SWITCHES

CHANCE VOUGHT AIRCRAFT
INCORPORATED · DALLAS, TEXAS

ORIGIN. BY A. R. Wilson GROUP UNIT 53725 REL. DATE ____ REL. DESK ____	ORIGIN. GP. APP. DATE DATE	[signature] PROJECT ENGINEER DATE DATE	 DATE DATE	AER-AVO-53720 CVA -0-182 PAGE 1 OF 7 ECP PC

TEST PROCEDURE

FOR

PACKAGE, MODULAR HYDRAULIC #2

ACTUATOR WITH MODULAR

END CAP MANIFOLD

CVS-54542-1

#2 PACKAGE
Test Procedure

1. SCOPE.-

1.1 Scope.- The Test procedure covers the testing requirements for the #2 modular hydraulic package. This package consists of the CVS-54542-1 Cyclinder Assembly and three modular valves; the Class A shuttle, one-way restrictor, and pressure relief valves.

1.2 Classification.- The #2 package is classified as a Class A package, indicating a flow range from 0 to 4 GPM.

2. APPLICABLE DRAWINGS.- The following drawings of the issue in effect on date of this test procedure form a part of the test procedure:

2.1 Drawings.-

CVS-54542	CVP-3344
CVS-54358	CVP-3346
CVS-54543	CVP-3350
CVS-54544	
CVS-54545	
CVS-54620	

3. TEST PROVISIONS.-

3.1 Test Fluid.- The test fluid shall be MLO-8200.

3.2 Test Fluid Temperature.- Tolerance on fluid temperatures specified herein shall be $\pm$ 5°F. The actuator outlet fluid temperature shall be specified, and the actuator inlet fluid temperature may be decreased a maximum of 25°F to compensate for heat generation. The fluid temperatures shall be measured as near as practicable to the outlet ports.

3.3 Filtration.- The test fluid shall be continuously filtered through a filter capable of removing all particles in excess of 25 microns (25 microns absolute), and shall remove 98% of all particles larger than 10 microns. The filter and element used shall be satisfactory for the temperature range encountered, and cleaned or changed regularly to prevent clogging. (Element shall conform to MIL-F-8815 testing procedures.)

3.4 Contamination.- No contaminating agents shall be purposely added to the test fluid.

3.5 Test Package.- The test package shall be partly or fully assembled, as dictated by the applicable paragraph, with provisions in the test set-up to cause each module to operate throughout its full cycle. All modules shall be checked for proper installation and safety wiring. The test package shall be set up per the applicable figure for each test phase.

4. TEST PROCEDURE.-

4.1 Module Function and Leakage Tests.-

4.1.1 Set up test apparatus as shown in Figure 1, with CVS-55087-1 and -2 plugs in place of the Piston and Rod Assembly and the Class A Restrictor, respectively. With test fluid at room temperature, pressure line selected to "alternate inlet," ports C_1 and C_2 open to atmosphere apply 4,000 psi hydraulic pressure. Maintain this pressure for five minutes. Leakage at port C_1 shall be negligible during this five minutes, leakage at port C_2 shall not exceed 90 drops per minute for the third minute. There shall be no appreciable leakage externally around modules.

4.1.2 Alter test set-up be selecting pressure line to port C_1. With test fluid at room temperature, port C_2 and "alternate inlet" port open to atmosphere, apply 4,000 psi hydraulic pressure. Maintain this pressure for five minutes. Leakage at "alternate inlet" port shall be negligible.

4.1.3 Repeat paragraphs 4.1.1 and 4.1.2, in that order, an additional 24 cycles.

4.1.4 Connect all lines to ports. Select pressure line to port C_2 and return line to port C_1, leaving "alternate inlet" port to vent. Shuttle in shuttle valve must be blocking the "alternate inlet" port. With test fluid at room temperature provide hydraulic pressure to port C_2 until flowmeter indicates a flow of 4 gpm. At this flow the differential pressure between gages G_2 and G_1 shall be approximately 4,900 psi, and G_1 shall read approximately 200 psi. Reduce pressure to zero and repeat cycle 24 times.

4.1.5 Repeat paragraphs 4.1.1 through 4.1.4 with test fluid at +450°F, with test fluid at -20°F.

4.1.6 Replace the CVS-55087-2 plug with a Class A restrictor, CVP-3344-2. With test fluid at -20°F maintain pressure line to port C_2 and return line to port C_1. Apply hydraulic pressure in 1 gpm increments until flowmeter reads 4 gpm. At this flow the differential pressure between gages G_2 and G_1 shall be approximately 2,050 psi, indicating the restrictor is passing flow at the proper pressure level. Record pressures at G_2 and G_1 for flows of 1, 2, 3 and 4 gpm.

4.1.7 Alter test set-up by selecting pressure line to port C_1 and return to port C_2. With test fluid at -20°F apply hydraulic pressure in 1 gpm increments until flowmeter reads 4 gpm. At this flow the differential pressure between gages G_1 and G_2 shall be at a minimum value. Record pressures at G_1 and G_2 for flows of 1, 2, 3 and 4 gpm.

4.1.8 Repeat paragraphs 4.1.6 and 4.1.7, in that order, an additional 24 cycles.

4.1.9 Repeat paragraphs 4.1.6 through 4.1.8 with test fluid at room temperature, with test fluid at +450°F.

4.2 Pressure Drop Tests.- In addition to the pressure drop testing of paragraphs 4.1.6 and 4.1.7 perform the following tests, with fluid at room temperature throughout.

4.2.1 Replace the Class A restrictor with the CVS-55087-2 plug. Select pressure line to port C_2 and return line to port C_1. Apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_2 and G_1 for these flow values.

4.2.2 Alter the test set-up by replacing the CVP-3350-1 Pressure Relief Valve with the Class A relief valve test cap. Maintain pressure line to port C_2 and return line to port C_1. Apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_2 and G1 for these flow values.

4.2.3 Alter the test set-up by replacing the Class A relief valve test cap with the CVP-3350-1 Pressure Relief Valve. Also replace the CVS-55087-2 plug with the Class A restrictor test cap. Maintain pressure line to port C_2 and return line to port C_1. Apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_2 and G_1 for these flow values.

4.2.4 Alter test set-up by selecting pressure line to port C_1 and return line to port C_2. Apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1 and G_2 for these flow values.

4.2.5 Alter test set-up by selecting pressure line to "alternate inlet" port, maintaining return line at port C_2. Apply sufficient hydraulic pressure to bottom the shuttle in the CVP-3346-1 Shuttle Valve against the C_1 port to prevent leakage to vent line. Reduce pressure to zero. Apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_3 and G_2 for these flow values.

4.3 Proof Pressure Tests.-

4.3.1 Set up test apparatus as shown in Figure 2, with all modules installed and the Piston and Rod Assembly installed in the CVS-54542-1 Actuator Assembly in place of the CVS-55087-1 Plug. Leave limit switches in-operative. With test fluid at room temperature, select pressure line to port C_1 and return line to port C_2. Place actuator in extended position. Apply hydraulic pressure at a rate not exceeding 25,000 psi per minute until pressure gage G_1 reads 6,000 psi $\pm$ 20 psi. Maintain this pressure for a minimum of two minutes. There shall be no evidence of excessive leakage, permanent set, or other damage to the package.

4.3.2 Alter test set-up by selecting pressure line to port C_2 and return line to port C_1. Slowly apply hydraulic pressure so as not to cause hard bottoming of the piston and increase pressure until pressure gage G_2 reads 6,000 psi $\pm$ 20 psi. Maintain this pressure for a minimum of two minutes. There shall be no evidence of excessive leakgee, permanent set, or other damage to the package.

4.3.3 Alter test set-up by selecting pressure line to the "alternate inlet" port and return line to port C_2. Place actuator in extended position. Apply hydraulic pressure at a rate not exceeding 25,000 psi per minute until pressure gage G_3 reads 6,000 psi $\pm$ 20 psi. Maintain this pressure for a minimum of two minutes. There shall be no evidence of excessive leakge, permanent set, or other damage to the package.

4.4 Cycling Tests.-

4.4.1 Set up test apparatus as shown in Figure 2, with limit switches set to permit the maximum permissible piston travel in extension and retraction without permitting piston bottoming. Limit switches shall operate the solenoid-operated selector valve. Utilize suitable actuator loading devices to control the rate of cycling. With test fluid at room temperature and the "alternate inlet" port vented to atmosphere supply 4,000 psi $\pm$ 20 psi hydraulic pressure. Regulate loading devices to permit approximately 24 cycles per minute. Operate at this pressure and rate for a minimum of five minutes. There shall be no evidence of excessive leakage or module mal-functioning.

4.4.2 Regulate loading devices to permit a flow rate of 4 gpm at 4,000 psi pressure. With test fluid at room temperature apply hydraulic pressure and cycle actuator for a minimum of two minutes. At two minutes lapsed time maintain pressure and increase flow rate to 8 gpm, cycling actuator for a minimum of one minute. There shall be no evidence of excessive leakage or module mal-functioning.

4.4.3 Select port C_1 open to atmosphere. Apply sufficient hydraulic pressure to bottom the shuttle in the CVP-3346-1 Shuttle Valve against the C_1 port to prevent leakage to vent line. Repeat paragraph 4.4.2.

4.4.4 With test fluid at room temperature, maintain port C_1 vented to atmosphere and supply 4,000 psi $\pm$ 20 psi hydraulic pressure. Regulate loading devices to permit approximately 24 cycles per minute. Operate at this pressure and rate for a minimum of five minutes. There shall be no evidence of excessive leakage or module mal-functioning.

4.4.5 Repeat paragraphs 4.4.1 through 4.4.4 with test fluid at +450°F.

FIGURE 1

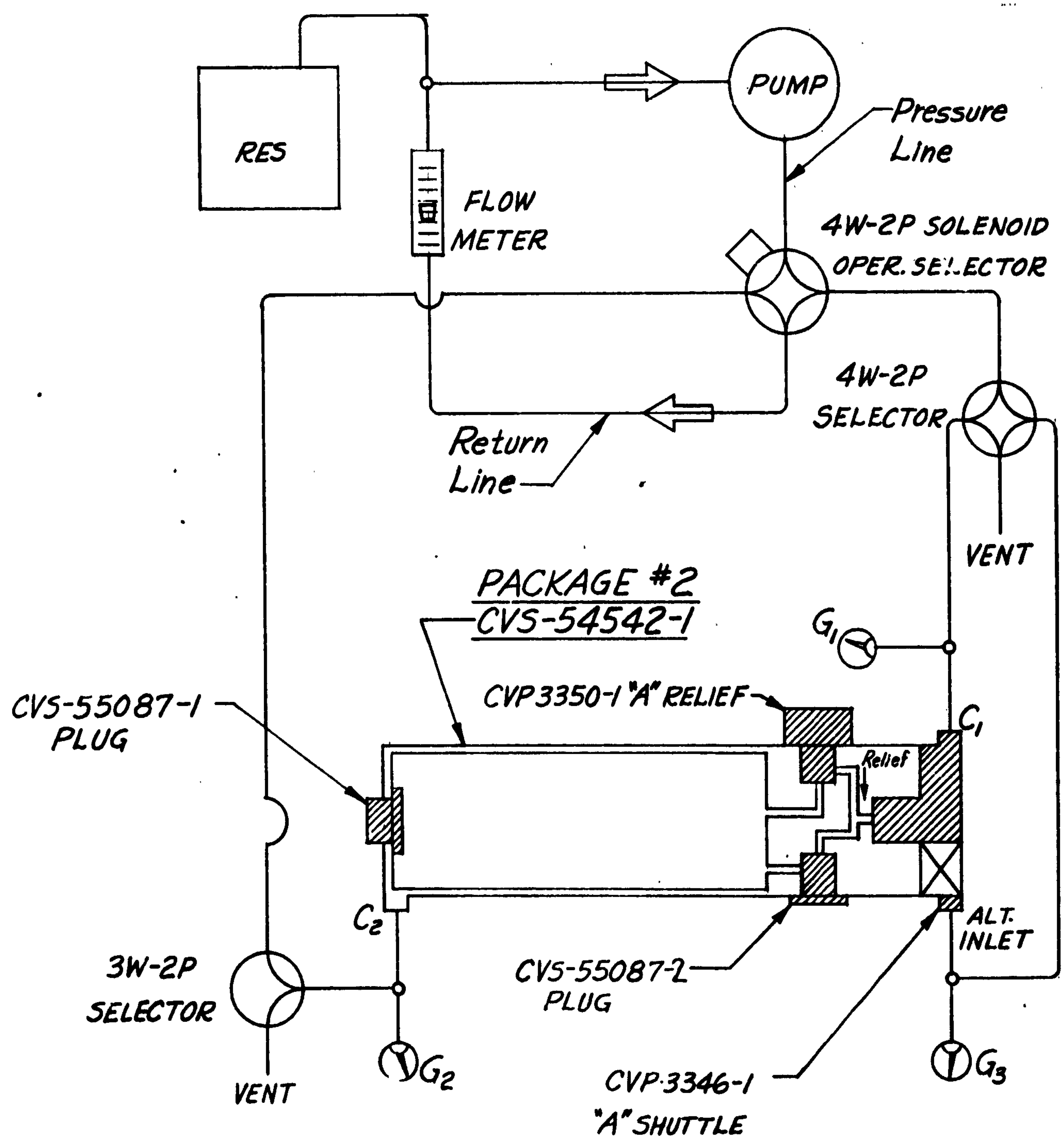
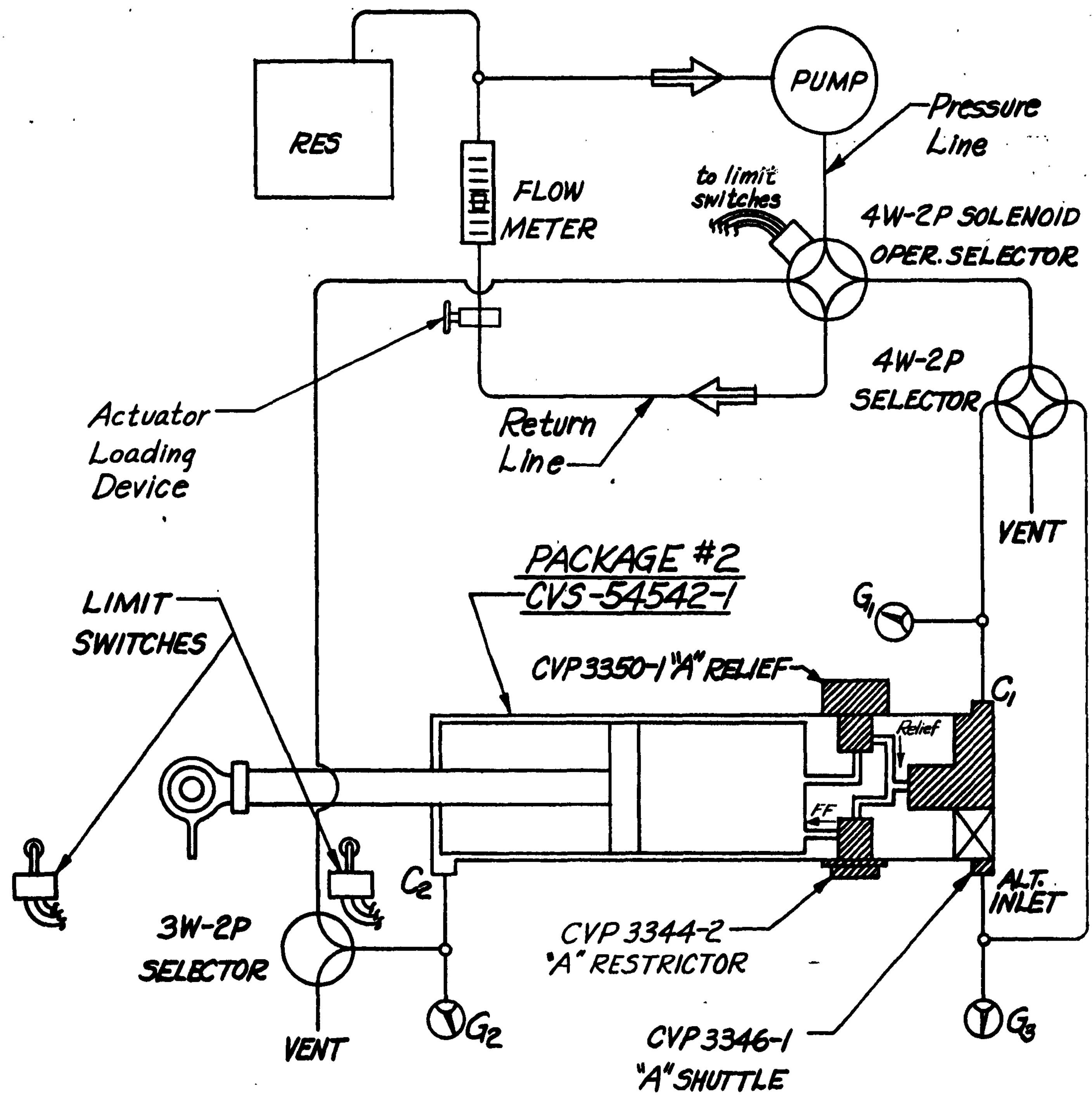

RES
PUMP
Pressure Line
FLOW METER
4W-2P SOLENOID OPER. SELECTOR
4W-2P SELECTOR
Return Line
VENT
PACKAGE #2
CVS-54542-1
G_1
CVS-55087-1 PLUG
CVP 3350-1 "A" RELIEF
C_1
Relief
C_2
ALT. INLET
3W-2P SELECTOR
CVS-55087-2 PLUG
VENT
G_2
CVP 3346-1 "A" SHUTTLE
G_3

FIGURE 2


RES
PUMP
Pressure Line
FLOW METER
to limit switches
4W-2P SOLENOID OPER. SELECTOR
4W-2P SELECTOR
Actuator Loading Device
Return Line
VENT
PACKAGE #2
CVS-54542-1
LIMIT SWITCHES
G1
CVP3350-1 "A" RELIEF
C1
Relief
FF
C2
ALT. INLET
3W-2P SELECTOR
CVP 3344-2
"A" RESTRICTOR
VENT
G2
CVP 3346-1
"A" SHUTTLE
G3

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AER-AVO-53720-0-183

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TEST PROCEDURE
FOR
PACKAGE, MODULAR HYDRAULIC #4
5 MODULE CONTROL
CVS-54926-1

CV

#4 PACKAGE
TEST PROCEDURE

1. SCOPE.-

1.1 Scope.- The test procedure covers the testing requirements for the No. 4 modular hydraulic package. This package consists of the CVS-54926-1 Manifold Assembly and five modular valves.

1.2 Classification.- The No. 4 package is classified as a Class A package, indicating a flow range from 0 to 4 GPM.

2. APPLICABLE DRAWINGS.- The following drawings of the issue in effect on date of this test procedure form a part of the test procedure:

2.1 Drawings.-

CVS-54926	CVP-3351
CVP-3348	CVP-3352
CVP-3349	CVP-3354

3. TEST PROVISIONS.-

3.1 Test Fluid.- The test fluid shall be MLO-8200.

3.2 Test Fluid Temperature.- Tolerance on fluid temperatures specified herein shall be $\pm 5^{\circ}F$. The outlet fluid temperature shall be specified, and the inlet fluid temperature may be decreased a maximum of $25^{\circ}F$ to compensate for heat generation. The fluid temperatures shall be measured as near as practicable to the outlet ports.

3.3 Filtration.- The test fluid shall be continuously filtered through a filter capable of removing all particles in excess of 25 microns (25 microns absolute), and shall remove 98% of all particles larger than 10 microns. The filter and element used shall be satisfactory for the temperature range encountered, and cleaned or changed regularly to prevent clogging. (Element shall conform to MIL-F-8815 testing procedures.)

3.4 Contamination.- No contaminating agents shall be purposely added to the test fluid.

3.5 Test Package.- The test package shall be partly or fully assembled, as dictated by the applicable paragraph, with provisions in the test set-up to cause each module to operate throughout its full cycle. All modules shall be checked for proper installation torque and safety wiring. The test package shall be set up per the applicable figure for each test phase.

4. TEST PROCEDURE.-

4.1 Module Function and Leakage Tests.-

4.1.1 Set up test apparatus as shown in Figure 1, with the limit switch and three-position switch arranged as shown. Energize solenoid #1 of CVP-3352 selector valve and energize CVP-3354 sequence valve. With actuators #1 and #2 in retracted position and test fluid at room temperature, apply hydraulic pressure until G_1 reads 1,000 psi $\pm$ 20 psi. Maintain this pressure for a minimum of 15 seconds. There shall be no movement of actuators #1 or #2 and there shall be no flow observed in the flowmeter. There shall be no appreciable leakage around modules.

4.1.2 With test set-up same as in paragraph 4.1.1 de-energize CVP-3354 sequence valve and increase hydraulic pressure until actuator #1 starts extension. At this point, pressure gage G_1 shall read approximately 1,500 psi. Maintain hydraulic pressure. Cylinder #1 shall extend fully and operate the limit switch. Limit switch actuation shall cause CVP-3354 sequence valve to become energized, permitting cylinder #2 to fully extend. Flow will be observed in flowmeter during cylinder #1 and #2 extension, at fully extended cylinders condition the flow shall drop to zero.

4.1.3 With no change to test set-up, increase hydraulic pressure until the CVP-3348 thermal relief valve relieves, causing the flowmeter to register a flow again. Pressure gage G1 shall read approximately 4,7000 psi when this flow begins. Decrease hydraulic pressure until the flowmeter indicates no flow. At this point pressure gage G_1 shall read between 820 psi and 1,800 psi. There shall be no external leakage around modules.

4.1.4 With both cylinders fully extended, de-energize solenoids #1 and #2. This shuts off all ports of CVP-3352 selector valve. Apply 4,000 psi $\pm$ 20 psi at pressure gage G_1 and hold this pressure for 15 seconds. There shall be no movement of the actuators and no flow indicated in the flowmeter.

4.1.5 With test fluid at room temperature, apply hydraulic pressure until pressure gage G_1 reads 2,000 psi $\pm$ 20 psi and energize solenoid #2 of CVP-3352 selector valve. Cylinder #2 shall fully retract and cylinder #1 shall remain extended. Increase hydraulic pressure until cylinder #1 actuates and moves to the fully retract position. At time cycinder #1 starts actuation, pressure gage G_1 shall read approximately 3,500 psi.

4.1.6 With both cylinders fully retracted de-energize solenoids #1 and #2. This shuts off all ports of CVP-3352 selector valve. Apply 4,000 psi $\pm$ 20 psi at pressure gage G_1 and hold this pressure for 15 seconds. There shall be no movement of the actuators and no flow indicated in the flowmeter.

4.1.7 Repeat paragraphs 4.1.1 thru 4.1.6 24 times.

4.1.8 Repeat paragraphs 4.1.1 thru 4.1.6 25 times with test fluid at +450°F.

4.1.9 Repeat paragraphs 4.1.1 thru 4.1.6 25 times with test fluid at -20°F.

4.2 Proof Pressure Tests.-

4.2.1 Alter test set-up of Figure 1 by removing actuators #1 and #2 and plugging the four open ports of the CVS-54926 manifold. With test fluid at room temperature, energize the CVP-3354 Sequence Valve, the #1 solenoid of CVP-3352 Selector Valve, and close the shut-off valve in the return line. Apply hydraulic pressure (hand pump optional) at a rate not exceeding 25,000 psi per minute until pressure gage G_2 reads 6,000 psi ± 20 psi. Maintain this pressure for a minimum of two minutes. There shall be no evidence of excessive leakage, permanent set, or other damage.to the leakage.

4.2.2 With test set-up same as in paragraph 4.2.1, keep CVP-3354 Sequence Valve energized and energize solenoid #2 of the CVP-3352 Selector Valve. Keep shut-off valve closed and with test fluid at room temperature apply hydraulic pressure at a rate not exceeding 25,000 psi per minute until pressure gage G_2 reads 6,000 psi ± 20 psi. Maintain this pressure for a minimum of two minutes. There shall be no evidence of excessive leakage, permanent set, or other damage to the package.

4.3 Pressure Drop Tests.-

4.3.1 Set up test apparatus in accordance with Figure 2 by removing plugs at the "lock" and "unlock" ports and installing jumper line with pressure gage G_3 between these ports. Maintain plugs at the "spread" and "fold" ports, energize the CVP-3354 Sequence Valve, and energize solenoid #1 of the CVP-3352 Selector Valve. With test fluid at room temperature apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flow meter. Record pressure readings at gages $G1$, G_2 and G_3 for these flow values.

4.3.2 Alter test set-up by energizing solenoid #2 of the CVP-3352 Selector Valve and de-energizing the CVP-3354 Sequence Valve. With test fluid at room temperature apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1, G_2 and G_3 for these flow values.

4.3.3 Alter the test set-up be replacing the CVP-3354 Sequence Valve with the sequence valve test cap. Repeat test of paragraph 4.3.2, disregarding sequence valve instructions in that paragraph.

4.3.4 Alter the test set-up by energizing solenoid#1 of the CVP-3352 Selector Valve. With test fluid at room temperature apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1, G_2 and G_3 for these flow values.

4.3.5 Alter test set-up of Figure 2 by installing the jumper line and pressure gage G_3 at the "spread" and "fold" ports and by plugging the "lock" and "unlock" ports. Energize the #1 solenoid of the CVP-3352 Selector Valve and with test fluid at room temperature, apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1, $G2$ and G_3 for these flow values.

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4.3.6 Alter test set-up by energizing solenoid #2 of the CVP-3352 Selector Valve. With test fluid at room temperature apply 4,000 psi ± 20 psi hydraulic pressure to assure full opening of the Class A Priority Valve. With this pressure, adjust flow to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1, G_2 and G_3 for these flow values.

NOTE: Shut-off valve in return line may be utilized to obtain the required pressure and flow settings.

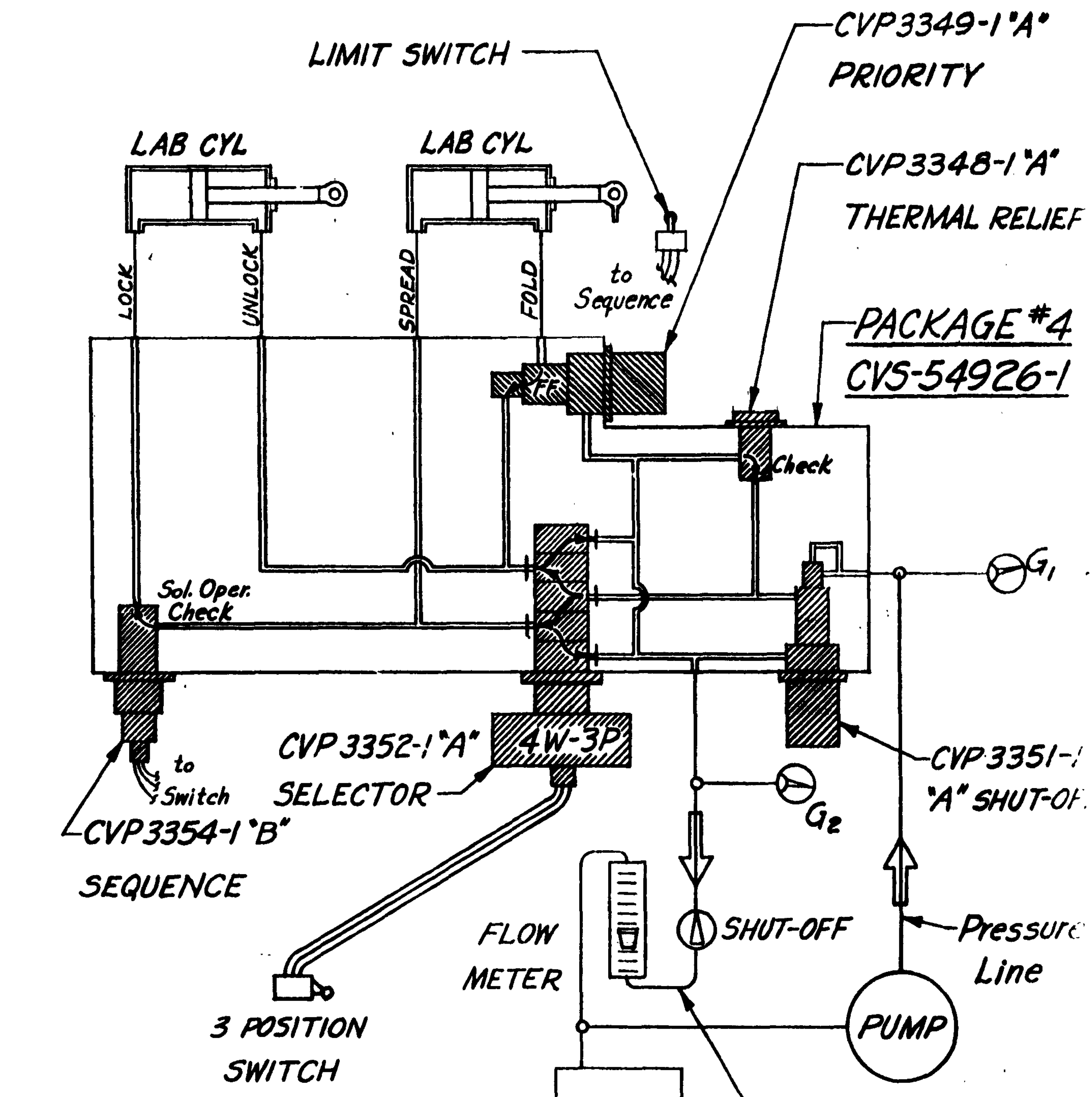
4.3.7 Alter test set-up by replacing the CVP-3349 Priority Valve with the priority valve test cap. With solenoid #2 of the CVP-3352 Selector Valve energized and test fluid at room temperature, apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1, G_2 and G_3 for these flow values.

4.3.8 Alter the test set-up by energizing solenoid #1 of the CVP-3352 Selector Valve. With test fluid at room temperature apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1, G_2 and G_3 for these flow values.

4.3.9 Alter the test set-up by removing jumper line and plug the "spread" and "fold" ports and de-energizing the #1 and #2 solenoids of CVP-3352 Selector Valve. With fluid at room temperature apply hydraulic pressure (will be necessary to exceed 4,700 psi approximately).to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1 and G_2 for these flow values.

4.3.10 Alter the test set-up be replacing the CVP-3348 Relief Valve with the Class A relief valve test cap. With test fluid at room temperature, apply hydraulic pressure to obtain 1, 2, 3 and 4 gpm flow at the flowmeter. Record pressure readings at gages G_1 and G_2 for these flow values.

FIGURE 1

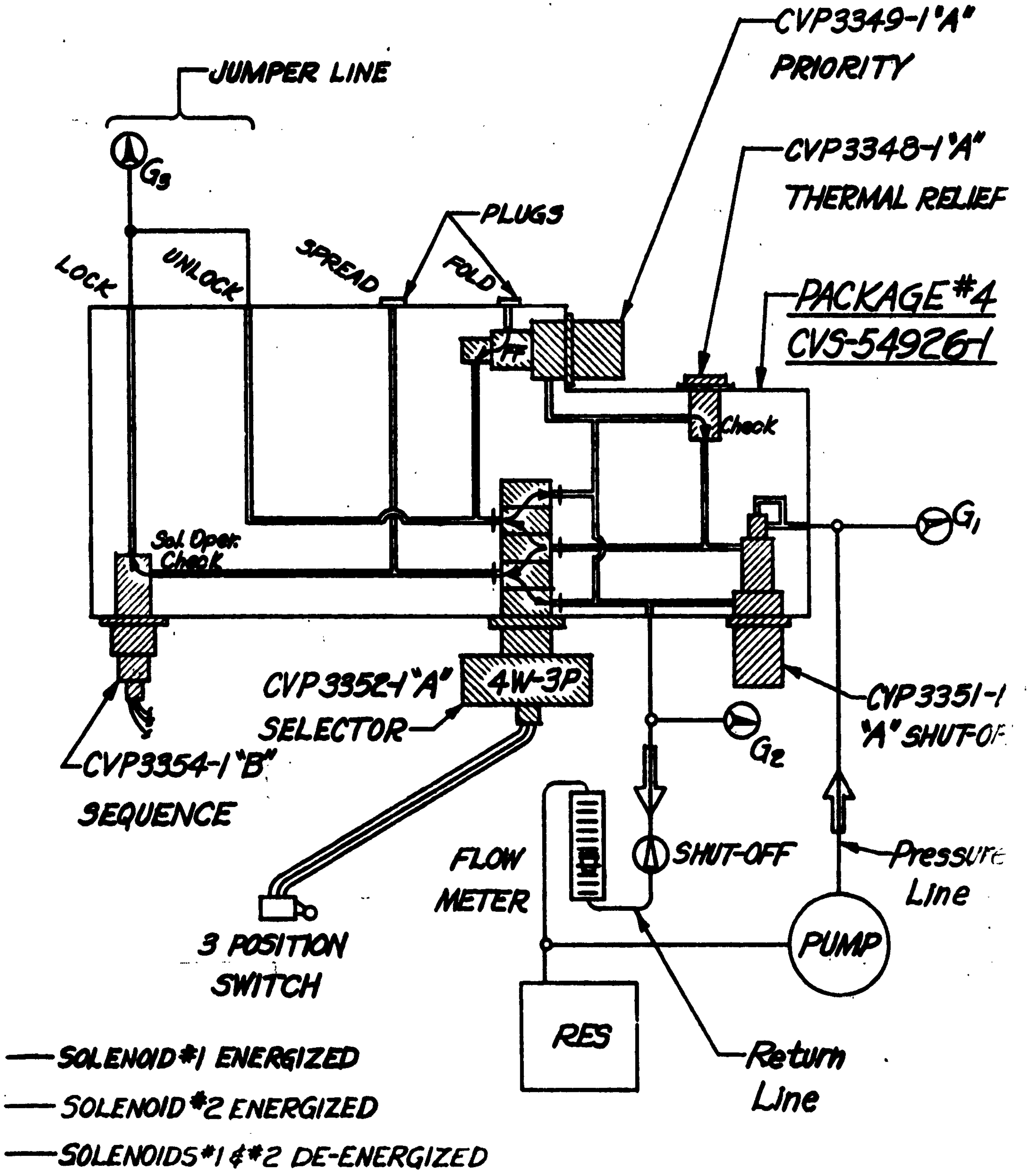


— SOLENOID #1 ENERGIZED

— SOLENOID #2 ENERGIZED

— SOLENOIDS #1 & #2 DE-ENERGIZED

FIGURE 2


JUMPER LINE
G_3
CVP3349-1 "A"
PRIORITY
CVP3348-1 "A"
THERMAL RELIEF
PLUGS
LOCK
UNLOCK
SPREAD
FOLD
PACKAGE #4
CVS-54926-1
Check
Sol. Oper. Check
G_1
CVP3352-1 "A"
4W-3P
SELECTOR
CVP3351-1
"A" SHUT-OF
CVP3354-1 "B"
SEQUENCE
G_2
FLOW
METER
SHUT-OFF
Pressure
Line
3 POSITION
SWITCH
PUMP
RES
Return
Line
SOLENOID #1 ENERGIZED
SOLENOID #2 ENERGIZED
SOLENOIDS #1 & #2 DE-ENERGIZED

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TEST PROCEDURE

FOR

MODULAR HYDRAULIC PACKAGE #5

CVS-54943-1 and CVS-54944-1

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ENGINEERING DEPARTMENT SPECIFICATION

#5 PACKAGE

TEST PROCEDURE

1. SCOPE -

1.1 Scope - The test procedure covers the testing requirements for the #5 modular hydraulic package. This package consists of the CVS-54943-1 and CVS-54944-1 manifolds, two filters, three modular valves and four automatic shut-off valves.

1.2 Classification - The #5 package is classified as a Class C package, indicating a flow range from 0 to 25 GPM.

2. APPLICABLE DRAWINGS - The following drawings of the issue in effect on date of this test procedure form a part of the test procedure:

2.1 Drawings -

CVS-54943	CVP-3347-1
CVS-54944	CVP-3350-2
CVS-54945	CVP-3353
CVS-54949	CVP-3356
CVS-54950	

3. TEST PROVISIONS -

3.1 Test Fluid - The test fluid shall be MLO-8200.

3.2 Test Fluid Temperature - Tolerance on fluid temperatures specified herein shall be ±5°F. The manifold outlet fluid temperature shall be specified, and the manifold inlet fluid temperature may be decreased a maximum of 25°F to compensate for heat generation. The fluid temperatures shall be measured as near as practicable to the outlet ports.

3.3 Filtration - The test fulid shall be continuously filtered through a filter capable of removing all particles in excess of 25 microns (25 microns absolute), and shall remove 98% of all particles larger than 10 microns. The filter and element used shall be satisfactory for the temperature range encountered, and cleaned or changed regularly to prevent clogging. (Element shall conform to MIL-F-8815 testing procedures.)

3.4 Contamination - No contaminating agents shall be purposely added to the test fluid.

3.5 Test Package - The test package shall be partly or fully assembled, as dictated by the applicable paragraph, with provisions in the test set-up to cause each module to operate throughout its full cycle. All modules shall be checked for proper installation and safety wiring. The test package shall be set up per Figure 1 for all tests.

4. Test Procedure

4.1 Pressure Drop Tests

4.1.1 Set up test apparatus per Figure 1, with the #5 package fully assembled with the exception of all automatic shut-off details. All four sets of shut-off detail parts (CVS-54945-1, -2, -3, -4 and -5, CVS-54949-1 and CVS-54950-1) shall not be installed in the manifolds until tests per paragraph 4.1.8 are to begin. Place the CVP3356 Solenoid-Operated Shut-Off Valve in the de-energized position, selector #1 in the blue position, selector #2 in the red position and the manual shut-off valve in the closed position. With test fluid at room temperature apply hydraulic pressure to obtain 5, 10, 15, 20 and 25 gpm flow at the flowmeter. Record pressures at gages G_1 and G_2 for these flow values. There shall be no appreciable leakage around filters, modular valves, or around common parting plane between manifolds.

4.1.2 Replace the CVP3353-3 Filter Assembly with the -3 filter assembly test cap and repeat pressure drop test of paragraph 4.1.1.

4.1.3 Alter test set-up by placing the CVP-3356 Solenoid-Operated Shut-Off Valve in the energized position and selector #1 in the red position. Maintain selector #2 in red position and the manual shut-off valve in the closed position. With test fluid at room temperature apply hydraulic pressure to obtain 3, 6, 9 and 12 gpm flow at the flowmeter. Record pressures at gages G_1 and G_Y for these flow rates. There shall be no appreciable leakage around filters, modular valves, or around common parting plane between manifold.

4.1.4 Replace the CVP-3356 Solenoid-Operated Shut-Off Valve with the Shut-Off Valve test cap and repeat pressure drop test of paragraph 4.1.3.

4.1.5 Replace the CVP-3353-2 Filter Assembly with the -2 Filter Assembly test cap and repeat pressure drop test of paragraph 4.1.3.

4.1.6 Maintain the three plugs in place of the CVP3353 -2 and -3 Filter Assemblies and the CVP-3356 Shut-Off Valve. Place selector #1 in the "all ports closed position" and the manual shut-off valve in the open position. With test fluid at room temperature apply hydraulic pressure to obtain 5, 10, 15, 20 and 25 gpm flow at the flowmeter. Record pressures at gages G_1 and G_3 for these flow values. There shall be no appreciable leakage around test plugs, modular valves, or around common parting plane between manifolds.

4.1.7 Replace the CVP-3350 Pressure Relief Valve with the Relief Valve test cap and repeat pressure drop test of paragraph 4.1.6.

4.1.8 Set up test apparatus per Figure 1, with all test plugs removed and all modules and filters re-installed. Install shut-off components in both manifolds. See CAUTION note below. Place the CVP-3356 Solenoid-Operated Shut-Off Valve in the de-energeized position, selector #1 in the blue position, selector #2 in the red position and the manual shut-off valve in the closed position. With test fluid at room temperature apply hydraulic pressure to obtain 5, 10, 15, 20 and 25 gpm flow at the flowmeter. Record pressures at gages G_1 and G_2 for these flow values. There shall be no appreciable leakage around filters, modular valves, or around common parting plane between manifolds.

CAUTION: Before installing shut-off components in manifolds, consult Modular Hydraulics Group for instructions.

4.1.9 Alter test set-up by placing the CVP-3356 Solenoid-Operated Shut-Off Valve in the energized position and selector #1 in the red position. Maintain selector #2 in the red position and the manual shut-off valve in the closed position. With test fluid at room temperature apply hydraulic pressure to obtain 3, 6, 9, and 12 gpm flow at the flowmeter. Record pressures at gages G_1 and G_4 for these flow values. There shall be no appreciable leakage around filters, modular valves, or around common parting plane between manifolds.

4.2 Module Function and Leakage Tests

4.2.1 With test apparatus set up as shown in Figure 1 place the CVP3356 Solenoid-Operated Shut-Off Valve in the de-energized position and selectors #1 and #2 in the blue position. Maintain the manual shut-off valve in the closed position and with test fluid at room temperature apply 4,000 psi ± 20 psi maximum at gage G_2. Actuator will operate between limit switches adjusted so as to prevent actuator bottoming during cycling. Operate actuator a minimum of 100 cycles. No appreciable leakage shall be permitted. Employ actuator loading device to regulate actuator to approximately 24 cycles per minute.

4.2.2 Alter test set-up by placing selector #1 in the dotted position (all ports closed) and with test fluid at room temperature apply hydraulic pressure until the CVP-3347 Pressure Switch operates the pressure indicator light. Pressure gage G_1 shall indicate approximately 2,700 psi when the indicator light operates. (This test assumes CVP-3347 Pressure Switch has been pre-set to operate at 2,700 psi ± 200 psi prior to installation in the manifold.) Reduce pressure to approximately 200 psi, indicator light shall be out. Repeat this cycle 24 times.

4.2.3 Alter test set-up by placing selector #1 in the red position and opening manual shut-off valve. Maintain selector #2 in the blue position and the CVP-3356 Solenoid-Operated Shut-Off Valve in the de-energized position. With test fluid at room temperature apply hydraulic pressure until the CVP-3350 Pressure Relief Valve permits the rated flow of 25 gpm to register in the flow meter. At rated flow of 25 gpm the differential pressure between gages G_1 and G_3 shall be approximately 4,850 psi. There shall be no appreciable leakage and no movement of the actuator. Reduce pressure at gage G_1 to approximately 500 psi. Repeat this cycle 10 times.

4.2.4 Alter test set-up by placing the CVP-3356 Solenoid-Operated Shut-Off Valve in the energized position and closing the manual shut-off valve. Maintain selectors #1 and #2 in the red and blue positions, respectively. With test fluid at room temperature apply 4,000 psi ±20 psi maximum at gage G_4. Actuator will operate between limit switches during cycling. Operate actuator a minimum of 100 cycles. No appreciable leakage shall be permitted. Employ actuator loading device to regulate actuator to approximately 24 cycles per minute.

4.2.5 Repeat paragraphs 4.2.1 through 4.2.4 with test fluid at +450°F.

4.2.6 Repeat paragraphs 4.2.1 through 4.2.4 with test fluid at -20°F.

4.2.7 Alter test set-up of Figure 1 as follows:

(a) Remove filter bowl and element of the CVP-3353-2 and CVP-3353-3 filter assemblies, being careful not to rotate the remaining part of the filters with respect to the CVS-54943 manifold.

(b) Disconnect the "cylinder pressure" and the "outlet pressure" lines from selector #1 and connect these lines to pump pressure line in accordance with dotted lines of Figure 1.

(c) Place selector #1 in dotted position, manual shut-off valve in closed position, and CVP 3356 shut-off valve in energized positions.

With test fluid at room temperature apply hydraulic pressure until gages G_1, G_2 and G_4 indicate 50 psi. Maintain this pressure for five minutes. Leakage shall be measured during the last three minutes, and shall not exceed 7 drops per minute for CVP-3353-2 and 8 drops per minute for CVP-3353-3. Increase hydraulic pressure to 4,000 psi ±20 psi and maintain this pressure for five minutes. Leakage shall be measured during the last three minutes, and shall not exceed 3 3/4 drops per minute for CVP-3353-2 and 4 1/3 drops per minute for CVP -3353-3.

4.2.8 Remove dotted pressure line and reconnect the "cylinder pressure" and "outlet pressure" lines to selector #1. Reinstall the CVP-3353-2 and -3 filter bowls with dummy elements made specifically to provide proper pressure drop across the pressure indicators. With test fluid at room temperature place the CVP-3356 shut-off valve in the de-energized position, manual shut-off valve closed, selector #1 in the blue position and selector #2 in the red position. Slowly apply hydraulic pressure. At some pressure between 70 and 90 psi, measured at G_1, the pressure indicator of the CVP-3353-3 filter shall be actuated and project from the filter bowl. Reduce hydraulic pressure to zero and manually re-set the pressure indicator. Repeat this test cycle 24 times.

4.2.9 Alter test set-up of paragraph 4.2.8 placing the CVP-3356 shut-off valve in the energized position and re-positioning selector #1 to the red position. Slowly apply hydraulic pressure. At some pressure between 70 and 90 psi, measured at G_1, the pressure indicator of the CVP-3353-2 filter shall be actuated and project from the filter bowl. Reduce hydraulic pressure to zero and manually re-set the pressure indicator. Repeat this test cycle 24 times.

4.3. PROOF PRESSURE TESTS

4.3.1 Remove the dummy elements from the filters and re-install actual filter elements. With test fluid at room temperature, manual shut-off valve in the closed position, CVP-335[illegible] shut-off valve in the ene[illegible]ized position and selector #1 in dotted position apply hy[illegible]ulic pressure (hand pum[illegible] [illegible]tional). Apply pressure at a rate not

exceeding 25,000 psi per minute until pressure gages G_1, and G_2, G_3 and G_4 read 6,000 psi ±20 psi. Maintain this pressure for a minimum of two minutes. There shall be no evidence of excessive leakage, permanent set, or other damage to the package.

4.3.2 Alter test set-up of paragraph 4.3.1 by de-energizing the CVP-3356 shut-off valve. Apply hydraulic pressure (hand pump optional) at a rate not exceeding 25,000 psi per minute until pressure gages G_1, G_2, and G_3 read 6,000 psi ±20. Maintain this pressure for a minimum of two minutes. Pressure gage G_4 shall indicate zero or near zero pressure. There shall be no evidence of excessive leakage, permanent set, or other damage to the package.

4.3.3 Set up test apparatus per Figure 2. The #5 package shall be in two parts, the CVS-54943 manifold assembly and the CVS-54944 manifold assembly. This paragraph is concerned with proof pressure testing of the shut-off features in the CVS-54944 manifold assembly only. Apply sufficient force by hand against shut-off pistons to cause each piston to unseat. Pistons shall reseat due to spring force of shut-off return springs. Operate each shut-off piston in this manner several times to assure unrestricted operation of each shut-off. Place the selector in the blue position and with test fluid at room temperature apply hydraulic pressure by hand pump, observing leakage past shut-off piston of pressure line for one minute at 5 psi, 100 psi, 500 psi, 1,000 psi and at proof pressure of 6,000 psi. Record leakage rate at each pressure setting read on pressure gage G_1. Reduce pressure to zero and hand-actuate the pressure line shut-off piston several times. Repeat the pressure tests at 5, 100, 500, 1000 and 6,000 psi and record leakage rates. Leakage rates shall not exceed 3 drops per minute at 5 psi nor 1 drop per minute at 6,000 psi.

4.3.4 Alter test set up of paragraph 4.3.3 by placing the selector in the red position and with test fluid at room temperature apply hydraulic pressure by hand-pump, observing leakage past the shut-off piston of the relief line for one minute at 5 psi, 100 psi, 500 psi, 1,000 psi and at proof pressure of 6,000 psi. Record leakage rate at each pressure setting read on pressure gage G_2. Reduce pressure to zero and hand-actuate the relief line shut-off piston several times. Repeat the pressure tests at 5, 100, 500, 1000 and 6,000 psi and record leakage rates. Leakage rates shall not exceed 3 drops per minute at 5 psi nor 1 drop per minute at 6,000 psi.

FIGURE 1

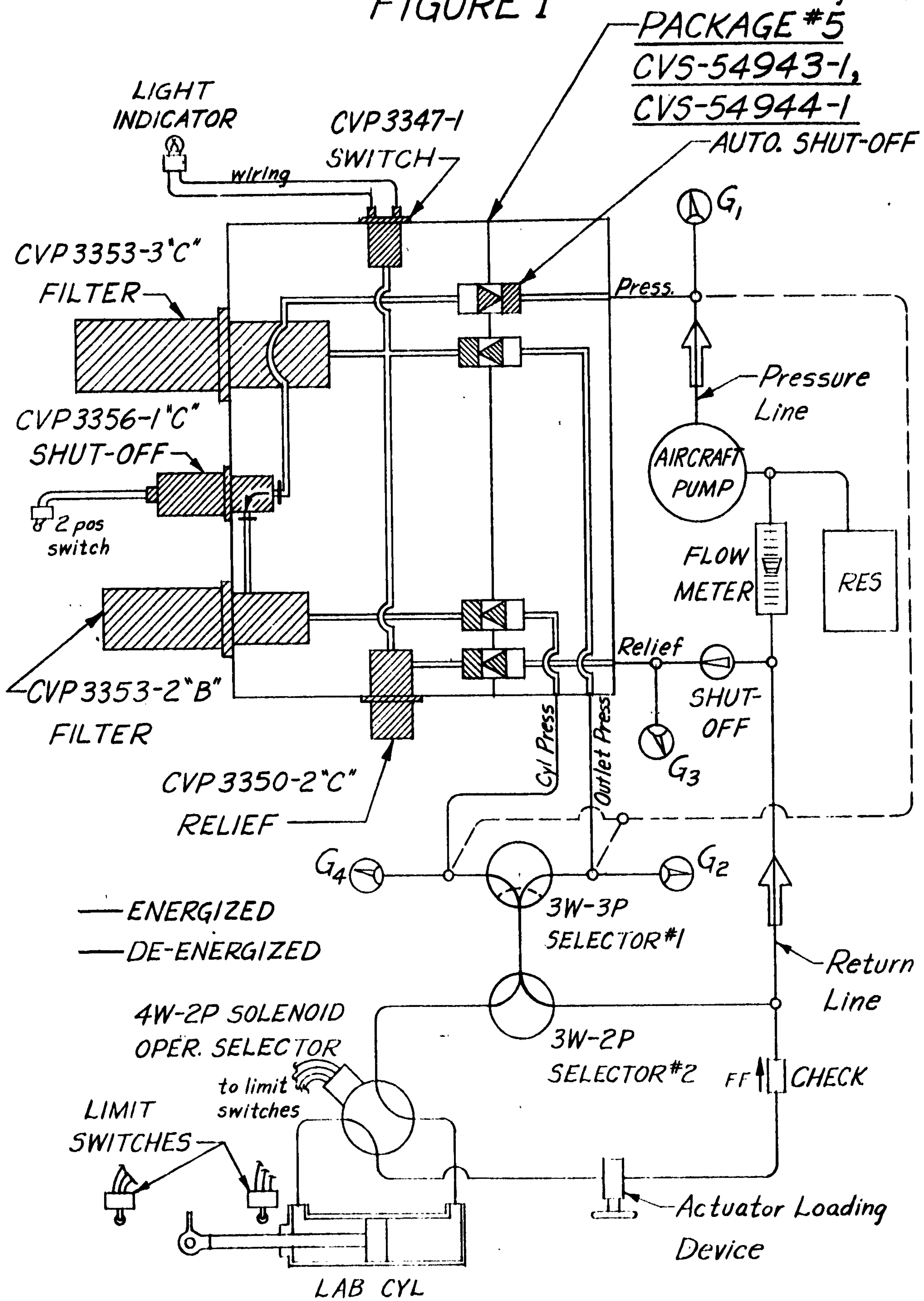
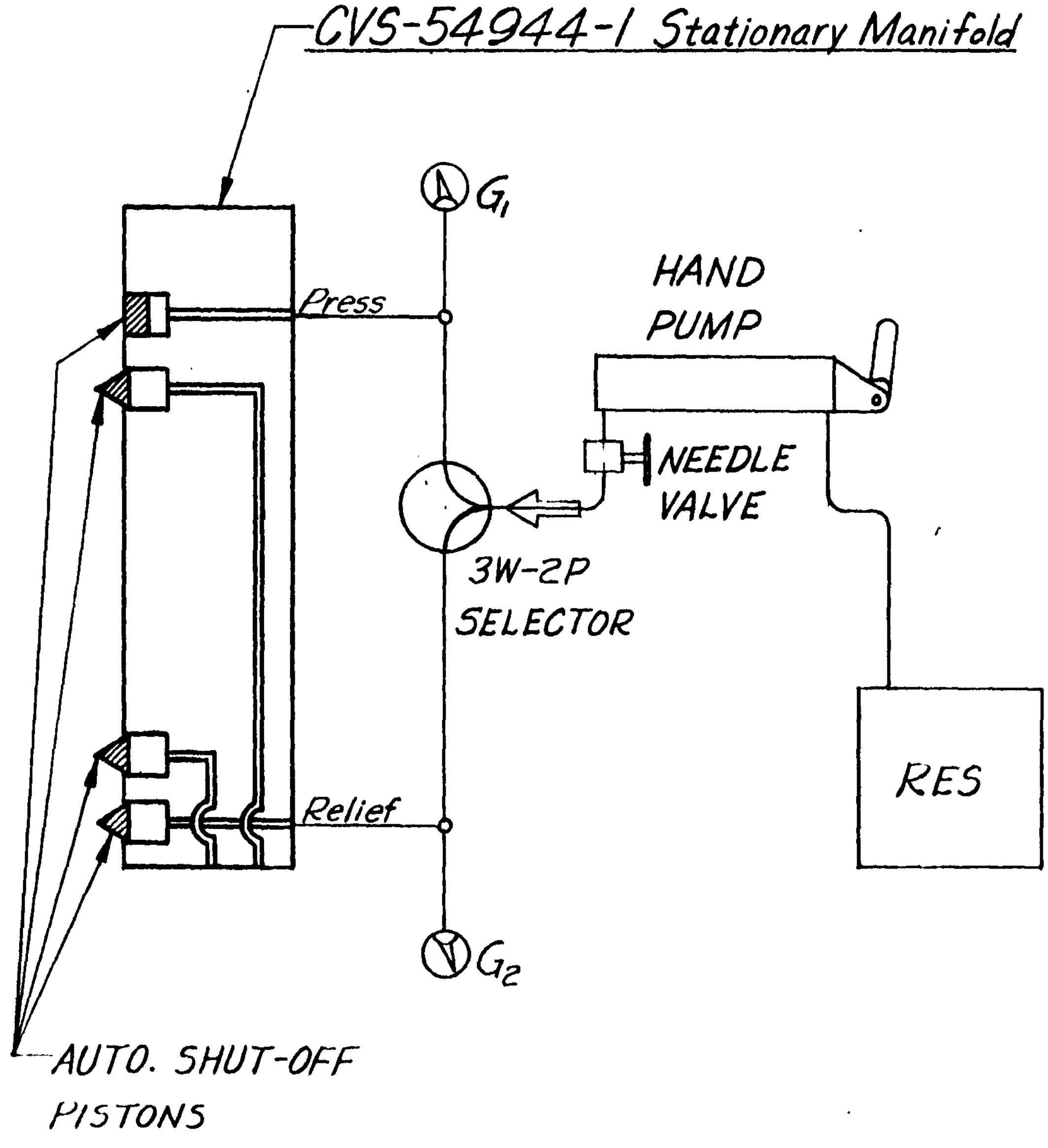

PACKAGE #5
CVS-54943-1,
CVS-54944-1
AUTO. SHUT-OFF
LIGHT
INDICATOR
wiring
CVP 3347-1
SWITCH
G_1
CVP 3353-3"C"
FILTER
Press.
Pressure
Line
CVP 3356-1"C"
SHUT-OFF
AIRCRAFT
PUMP
2 pos
switch
FLOW
METER
RES
CVP 3353-2"B"
FILTER
Relief
SHUT-
OFF
G_3
Cyl Press
Outlet Press
CVP 3350-2"C"
RELIEF
G_4
G_2
3W-3P
SELECTOR #1
ENERGIZED
DE-ENERGIZED
Return
Line
4W-2P SOLENOID
OPER. SELECTOR
3W-2P
SELECTOR #2
FF
CHECK
to limit
switches
LIMIT
SWITCHES
Actuator Loading
Device
LAB CYL

FIGURE 2


CVS-54944-1 Stationary Manifold
G_1
Press
HAND
PUMP
NEEDLE
VALVE
3W-2P
SELECTOR
RES
Relief
G_2
AUTO. SHUT-OFF
PISTONS

APPENDIX III-3

LEE PLUG AND STRATOFLEX PLUG TESTS

APPENDIX III-3

LEE PLUG AND STRATOFLEX PLUG TESTS

LEE PLUG TESTS

Twelve Lee plug test samples divided into three sizes were evaluated. The purpose of this plug is to externally seal drilled passages that are made in the manifold blocks during manufacturing.

The assembled unit consists of two parts: a plug and a pin as shown in Figure III-28 . The plug has minute annular lands on the O.D. and a slightly tapered hole. The pin has a matching taper and as the pin is driven into the tapered hole of the plug, the O.D. expands into the bore of the manifold. This expansion pressure is of such magnitude that the manifold material brinnels and moulds around the annular lands of the plug, thereby effecting a seal and preventing blowout.

Two each of a -2S, -4S and -7S Lee plug were tested. The sleeves and plugs were assembled into a manifold containing ports designed according to the vendor's specifications. Pertinent plug and port data are given in Table III-4. Static hydraulic pressures to 6,000 psi at room temperature were applied to the plugs with no leakage occurring. The plugs and manifold were then installed in a temperature box and subjected to a total of 47,400 pressure impulse cycles to 6,000 psi while being subjected to temperatures ranging from -65°F to 450°F. There was no evidence of leakage at any time throughout the tests.

Use of Lee plugs as a passage seal appears very encouraging with one exception. Forces required to insert the pin into the plug were much higher than expected, ranging from 1,800 pounds to 13,020 pounds for the smallest to the largest sizes, respectively. All pins were lubricated with either Oronite 8200 hydraulic fluid or with MIL-G-3278A grease before insertion. Pins were installed and the forces were applied with a compression test machine.

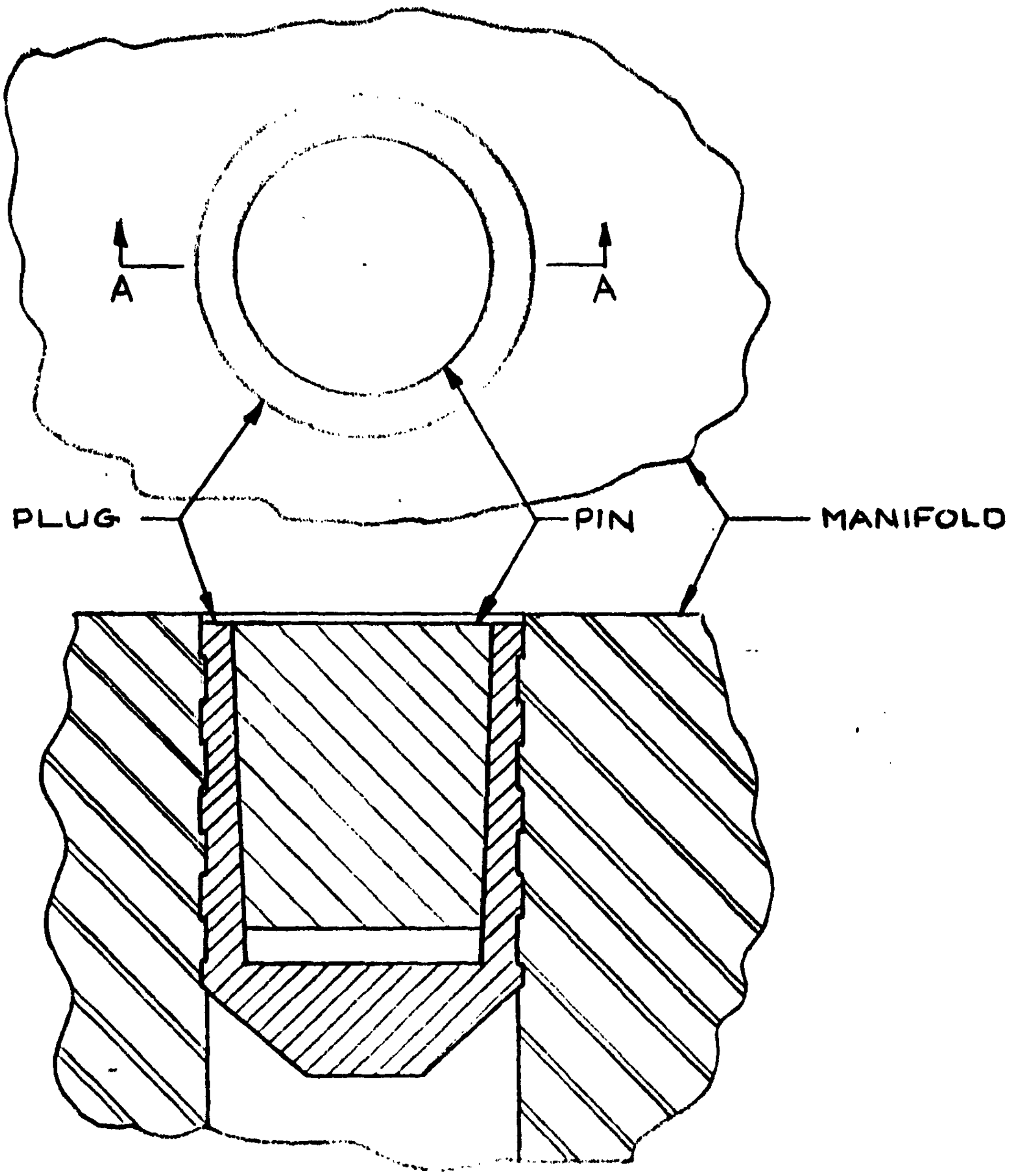
Additional tests were performed on the Lee plugs to determine the significance of the high installation forces previously found. Other tests were performed to determine seal performance when installed in thin wall ports.

Two each of a -2S, -4S and -7S Lee pin and plug were installed in a common test housing having ports designed according to the vendor's specifications. Pertinent plug, port and test data are given in Table III-5 . The plugs were proof pressure tested at 450°F and 6,000 psi and were subjected to 50,000 p.i.c. (pressure impulse cycles) in accordance with the pressure-time-temperature spectrum presented in Figure II-22. Upon completion of the pressure impulse cycling, a burst pressure test of 10,000 psi was performed. No leakage or failure occurred at any time during the testing.

One each of a -2S and a -4S plug design, both S/N 5, was installed into a thin wall port made by machining off the threads and boring out the I.D.

FIGURE III-28

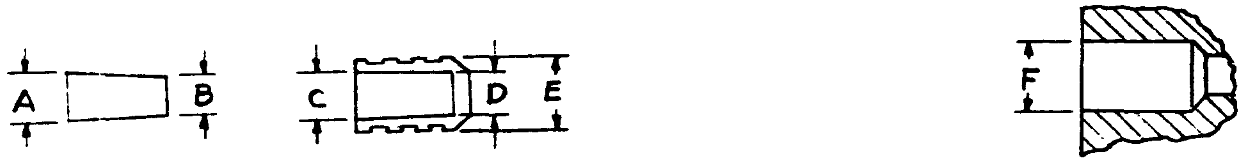
LEE PLUG



SECTION AA

TABLE IE-4

LEE PLUG AND PORT DATA



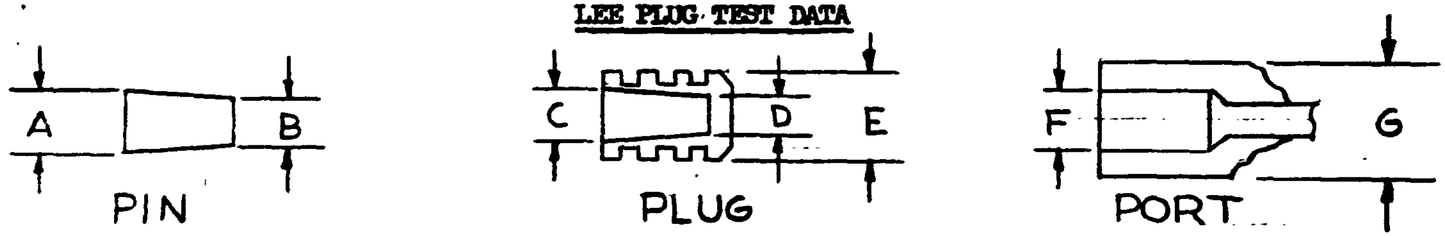
PIN PLUG PORT

DASH NO.	LAB S/N	A (IN.)	B (IN.)	C (IN.)	D (IN.)	E (IN.)	F (IN.)	Insert Force (LB)	Lubricant Used
2S	1 2	.1805 .1805	.1762 .1765	.1775 .1770	.1690 .1690	.2179 .2178	.219 .219	2,200 1,800	Oronite 8200 MIL-G-3278A
4S	1 2	.3075 .3075	.3010 .3010	.3015 .3015	.2954 .2954	.3429 .3430	.344 .345	3,380 1,940	Oronite 8200 MIL-G-3278A
7S	1 2	.4960 .4953	.4878 .4878	.4900 .4905	.4801 .4802	.5302 .5305	.531 .531	11,160 13,020	Oronite 8200 MIL-G-3278A

NOTE: Port material was CRES 17-4PH aged to Rockwell C40-C46.

TABLE III-5

LEE PLUG TEST DATA



DASH NO	LAB S/N	A-DIA (IN.)	B-DIA (IN.)	C-DIA (IN.)	D-DIA (IN.)	E-DIA (IN.)	F-DIA (IN.)	G-DIA (IN.)	RAPS* To Install	RESULTS PRES. IMPULSE CYCLING
2 s	3	.1803	.1761	.1775	.1685	.2179	.2191	**	8	50,000 p.i.c. without leakage
	4	.1804	.1765	.1775	.1695	.2180	.2191	**	8	" " " "
	5	.1804	.1763	.1780	.1688	.2180	.2189	.3571	8	" " " "
4 s	3	.3074	.3010	.3013	.2953	.3430	.3441	**	12	50,000 p.i.c. without leakage
	4	.3076	.3020	.3013	.2952	.3429	.3441	**	12	" " " "
	5	.3075	.3010	.3016	.2955	.3429	.3442	.4241	12	Blew out after 8,667 p.i.c.-No prior leaks
7 s	3	.4950	.4875	.4905	.4818	.5305	.5320	**	15	50,000 p.i.c. without leakage
	4	.4949	.4875	.4900	.4819	.5305	.5320	**	15	" " " "

NOTES: 1. *One rap consisting of delivering one sharpblow to the pin by hand with a 2 lb. brass mallet from a height of approximately 15 inches. In previous similar test installations where the plugs were pressed in by a hydraulic press, forces up to 13,000 lbs were required for pin insertion.

2. ** All ports so indicated were in a common test housing 2.25 x 2.50 x 6.0 inches made of CRES 17-4PH heat treated to a hardness of RC 40-46.

3. All plugs and pins were lubricated with MLO-8200 Hydraulic Fluid before installation.

of two AN stainless stub fittings of appropriate size. Pertinent plug, port and test data are shown in Table III-5. The -2S plug design completed a proof pressure test of 50,000 p.i.c. and a 10,000 psi burst pressure test without leakage or failure. The -4S plug blew out of the fitting after 8,667 p.i.c. Visual examination of the port and plug indicated that little or no upsetting of metal had occurred when the plug was originally installed. The -4S plug with the pin was coated with Biggs bonding agent, #385, re-installed in the port and baked at 450°F for 1 hour. The force required for this installation was approximately the same as that of the first installation. The plug and port fitting were installed in the temperature box and p.i.c. was resumed simultaneously with raising of the temperature. After 1,358 p.i.c. and at an oil temperature of 405°F, the plug again blew out.

Test results indicate that the Lee plug is a reliable means of plugging port holes provided: (1) the port has a reasonable wall thickness (minimum of approximately 1/3 of plug diameter), (2) port tolerances are accurately controlled, and (3) port material is sufficiently hard.

STRATOFLEX PLUG TEST

Twelve Stratoflex plug test samples divided into four sizes were evaluated. The purpose of this plug is to externally seal drilled passages that are made in the manifolds during fabrication. The assembled unit consists of a solid threaded plug screwed into a flareless type manifold port as shown in Figure III-29.

Two each of a -3, -5, -8 and -10 Stratoflex plug, as described in the previous Quarterly Progress Report, were tested. The plugs were installed in a single manifold containing ports as described in Table III-6 . Physical data for the plugs, initial torque and maximum torque imposed on the plugs before being imposed to impulse pressure cycling are also shown. Static pressure sealing at room temperature was achieved on all plugs with the exception of the -8 S/N1 plug. Leakage for this plug was very low at 6,000 psi and was considered sufficiently low so as to allow the performance of pressure impulse cycling.

With the plugs tightened to the maximum torques shown in Table III-6 the plugs were installed in the temperature box and subjected to 47,400 p.i.c. at temperatures from -65°F to 450°F with no evidence of leakage or with no increase of leakage in the case of the -8 plug in the temperature range from 75°F to 450°F. At a temperature of -65°F, all plugs were found to leak significantly under pressure impulse cycling. Upon raising the temperature to room temperature and above, the plugs were again found to seal as before.

The manifold was then removed from the temperature box. One plug (those designated S/N1) of each size was removed and the sealing surfaces of the -5, -8, -10, and the threads of the -3 plug, were coated with a bonding agent. The bonding agent was No. 385 from Carl H. Biggs Company, 2255 Barry Avenue, Los Angeles 64, California. After application of the agent, the plugs were installed in the manifold and baked at 375 ± 20°F for 1 hour.

After baking, the plugs were subjected again to static hydraulic pressures. Only the -8 S/N1 plug was found to be leaking. An effort was made to remove the plug, however it could not be removed without stripping the internal wrenching socket.

The poor sealing performance of the plugs is no doubt due in part to the ports of the manifold being slightly out of tolerance. The ports were machined on an individual basis rather than being finished with a single tool containing the port profile. The latter method would eliminate all spiraling and enable the forming of closer tolerance ports.

FORM NO. 38

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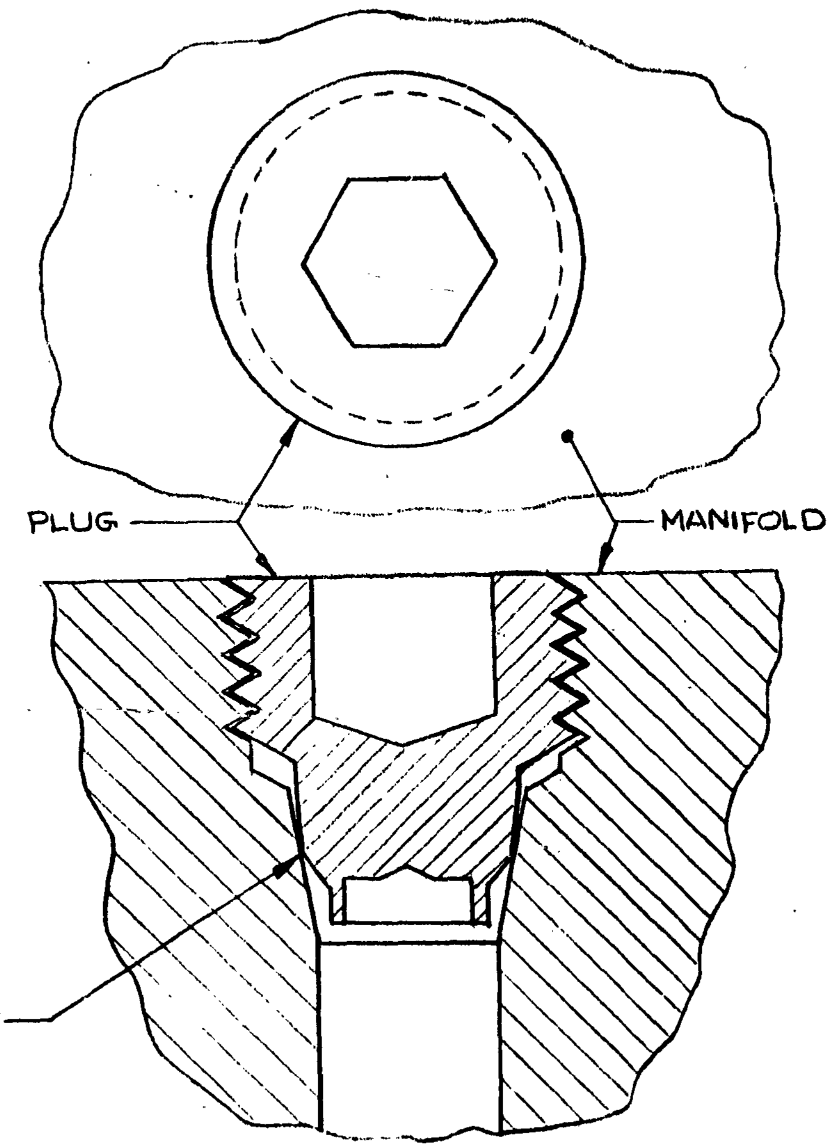
FIGURE III-29

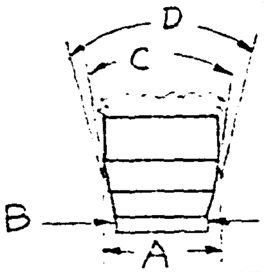
STRATOFLEX PLUG



FORM NO. 38

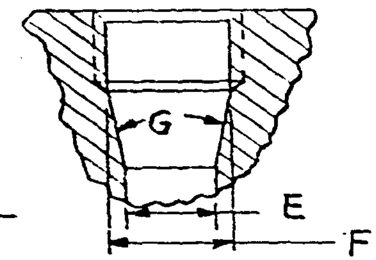
TABLE III-6

STRATOFLEX PLUG AND PORT DATA



PLUG

PORT



DASH NO.	S/N	A (Inches)	B (Inches)	C (Degree)	D (Degree)	E* +.004 -.000 (Inches)	F* +.004 -.000 (Inches)	G (Degree)	Torque (In/lb) Min.	Max.	NOTES
-3	Req'd	-	-	-	-	.196	.267	24±2	-	-	-
	1	.267	.186	12:55	37:03	.194	.251	23:24	~~95~~	95	3,4
	2	.265	.186	12:52	37:15	.195	.270	24:51	95	**	3,
-5	Req'd	-	-	-	-	.324	.382	24±2	-	-	-
	1	.380	.313	13:10	41:50	.325	.379	29:47	190	350	3,4
	2	.388	.315	12:58	39:28	.327	.363	30:22	190	300	3,
-8	Req'd	-	-	-	-	.514	.601	24±2	-	-	-
	1	.582	.501	13:33	37:52	.516	.585	28:20	475	1200	3,5
	2	.587	.500	14:12	37:54	.514	.599	26:56	475	1000	3,
-10	Req'd	-	-	-	-	.641	.727	24±2	-	-	-
	1	.719	.625	12:54	37:15	.641	.701	29:48	675	1500	3,4
	2	.719	.624	13:01	37:13	.639	.712	29:40	675	1500	3,

NOTES:
1. *Tolerances apply to required valves only.
2. **In raising torque, internal wrench socket was stripped out between 150 and 200 in/lbs.
3. Plugs leaked at -65°F.
4. These plugs did not leak under static pressure after applying Biggs Bonding Agent No. 385.
5. This seal leaked under static pressure with bonding agent.
6. Port material was CRES 17-4PH aged to Rockwell C40 - C46.

APPENDIX III-4

COMPONENT LOCKING DEVICES

APPENDIX III-4

COMPONENT LOCKING DEVICES

HELI-COIL INSERT TEST

Preliminary tests were conducted on the locking Heli-Coil insert to determine the torque characteristics of this type insert which has the third coil deformed to provide ten flat sections in the helix. These flat sections provide interference with the thread installed in the insert, thereby providing a locking action. The torque required to thread the test plug into the test block in which the insert was installed was recorded every quarter turn from engagement of the first deformation until the shoulder of the plug bottomed on the block. Readings were also taken every quarter turn as the plug was removed from the block. These values were taken for the following conditions:

1. The original insert with retaining roll pin still in place

2. The original insert with retaining roll pin removed

3. A second insert without a retaining pin.

The plug was installed and removed four times in each of the above conditions. The data for the installations is presented in Figures III-30.

The torques required for removal are comparable to those for installation and in some instances even higher. So far, indications are that repeated installations will lower the torque values for installation and removal. Motion of the insert in the test block was not detected after removal of the retaining pin from the original insert nor during the second insert test. However, the plug has only three complete threads which are not enough to engage all the deformations of the insert.

The results of these preliminary tests warranted further investigation of the capabilities of this type of thread locking device. In preparation for more extensive tests, a manifold which utilizes a 1-3/8 - 12 UNF Heli-Coil screw-lock insert and a plug of the same thread size were designed. Two plugs were fabricated and two manifolds were partially completed. These manifolds were sent to the vendor for tapping with an insert tap and installation of the Heli-Coil inserts.

The two manifolds which had been sent to the vendor for installation of screw-lock inserts were received. The plug, S/N 1, and manifold, S/N 1, were assembled. The torque required to install the plug was recorded at each quarter turn. The assembly was proof pressure tested for 2 minutes at 6,000 psi. The assembly was then installed in the temperature box of the impulse cycling system and tested to the following conditions: hydraulic impulse cycle from zero pressure to a peak surge pressure of 6,000 psi, then to a system operating pressure of 4,000 psi, and back to zero pressure. The cycling rate was 35 cpm and the fluid and ambient temperatures were 450°F

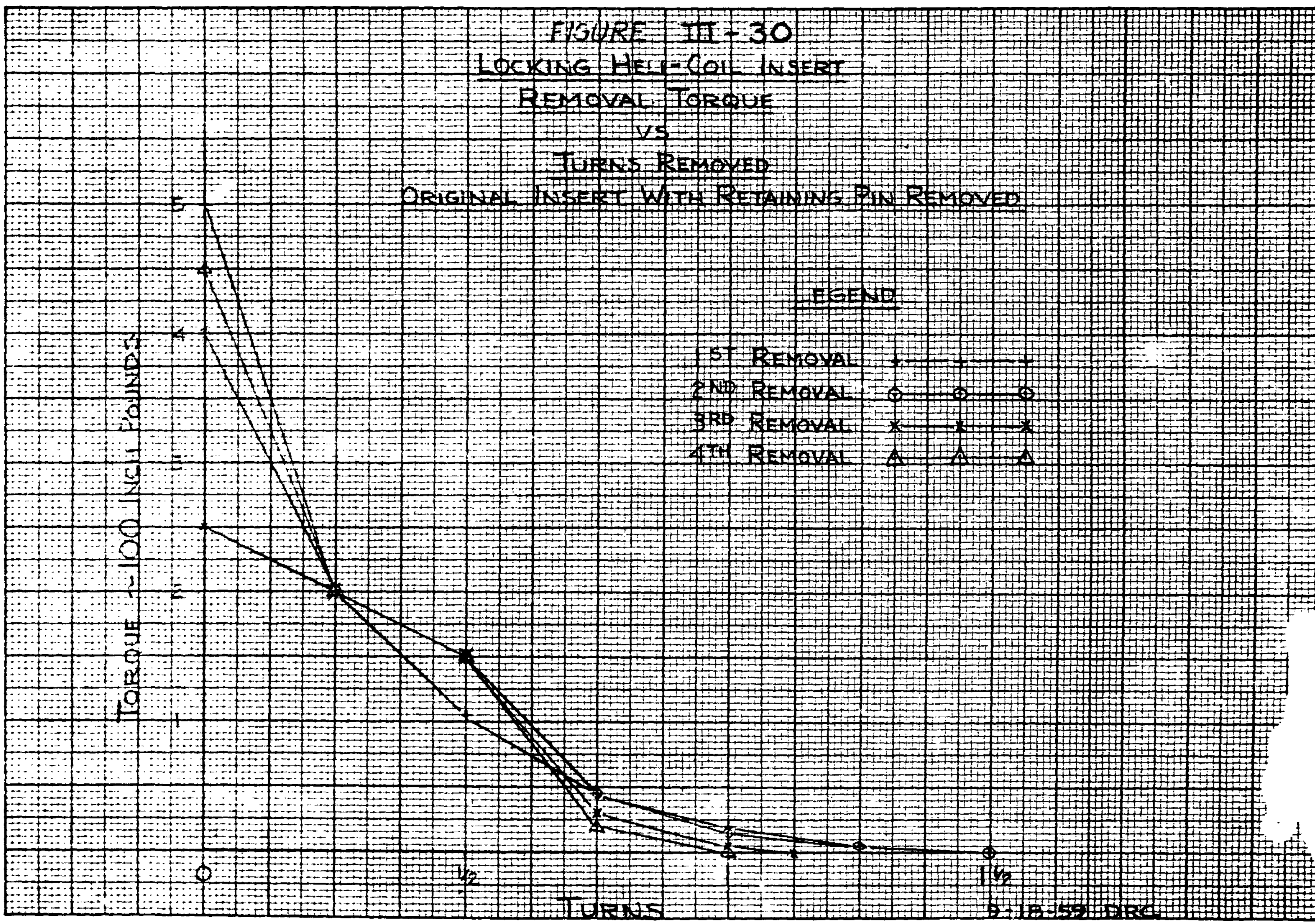


FIGURE III-30
LOCKING HELI-COIL INSERT
REMOVAL TORQUE
VS
TURNS REMOVED
ORIGINAL INSERT WITH RETAINING PIN REMOVED
LEGEND
1ST REMOVAL
2ND REMOVAL
3RD REMOVAL
4TH REMOVAL
TORQUE ~ 100 INCH POUNDS
5
4
3
2
1
0
1/2
1 1/2
TURNS
9-18-59 DRC

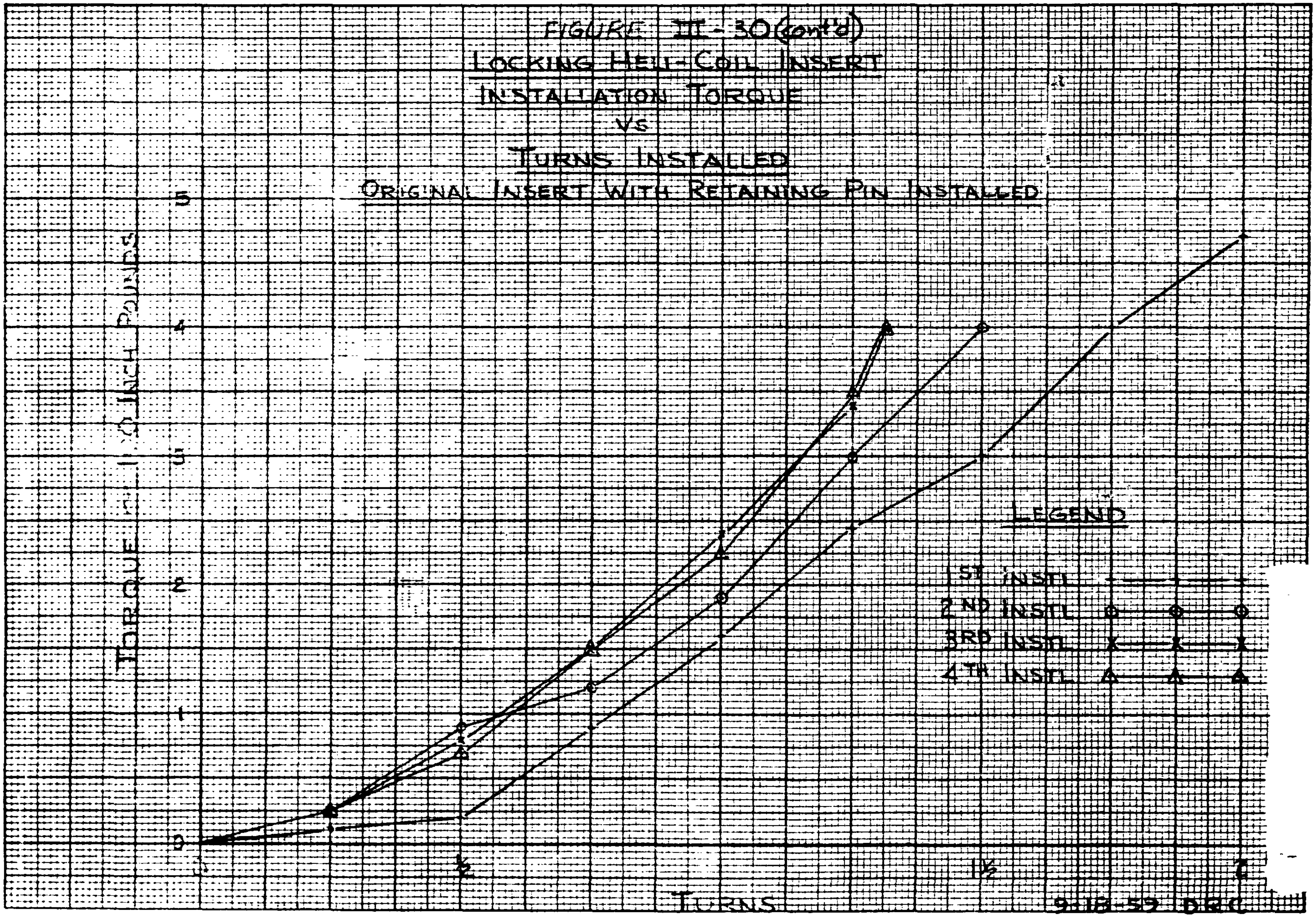


FIGURE III-30 (cont'd)
LOCKING HELI-COIL INSERT
INSTALLATION TORQUE
VS
TURNS INSTALLED
ORIGINAL INSERT WITH RETAINING PIN INSTALLED
TORQUE IN INCH POUNDS
0
1
2
3
4
5
½
1½
2
TURNS
LEGEND
1ST INSTL
2ND INSTL
3RD INSTL
4TH INSTL
9-18-59 DRC

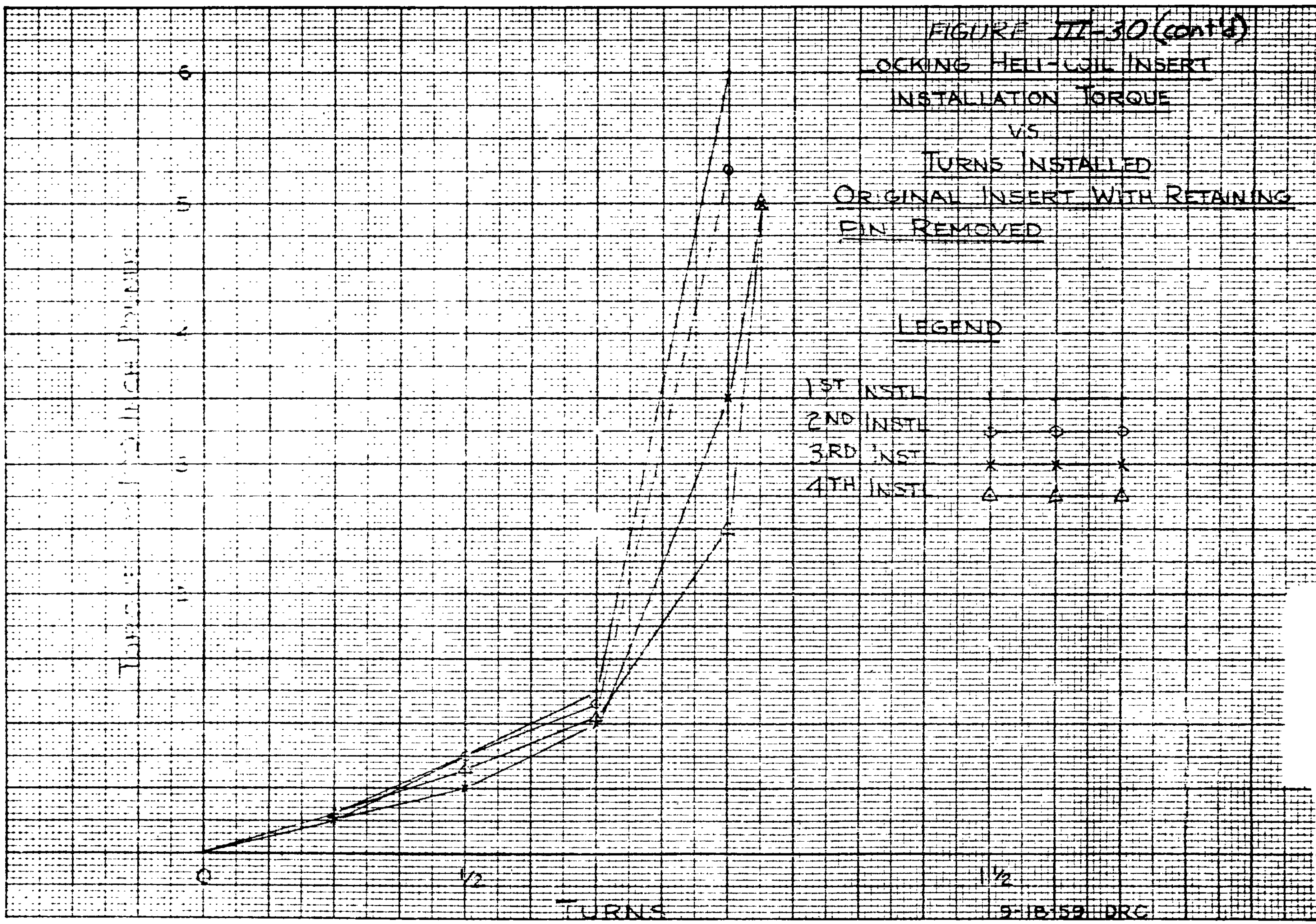


FIGURE III-30 (cont'd)
LOCKING HELI-COIL INSERT
INSTALLATION TORQUE
VS
TURNS INSTALLED
ORIGINAL INSERT WITH RETAINING
PIN REMOVED
LEGEND
1ST INSTL
2ND INSTL
3RD INSTL
4TH INSTL
6
5
0
1/2
1 1/2
TURNS
9-18-59 DRC

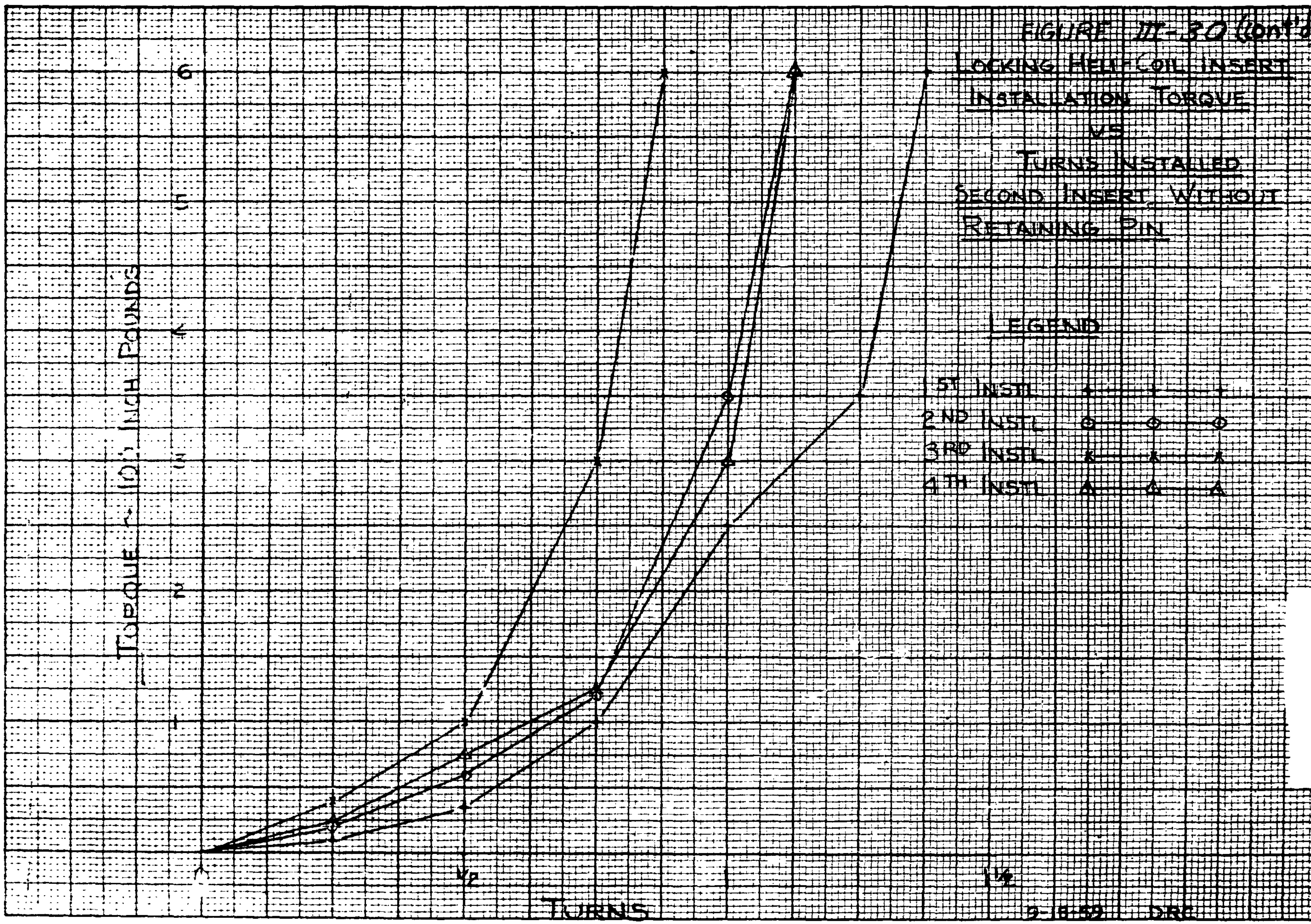


FIGURE III-30 (Cont'd)
LOCKING HELI-COIL INSERT
INSTALLATION TORQUE
VS
TURNS INSTALLED
SECOND INSERT WITHOUT
RETAINING PIN
LEGEND
1ST INSTL
2ND INSTL
3RD INSTL
4TH INSTL
TORQUE ~ IN. INCH POUNDS
6
5
4
3
2
1
0
1/2
1
1 1/2
TURNS
9-18-59 DRC

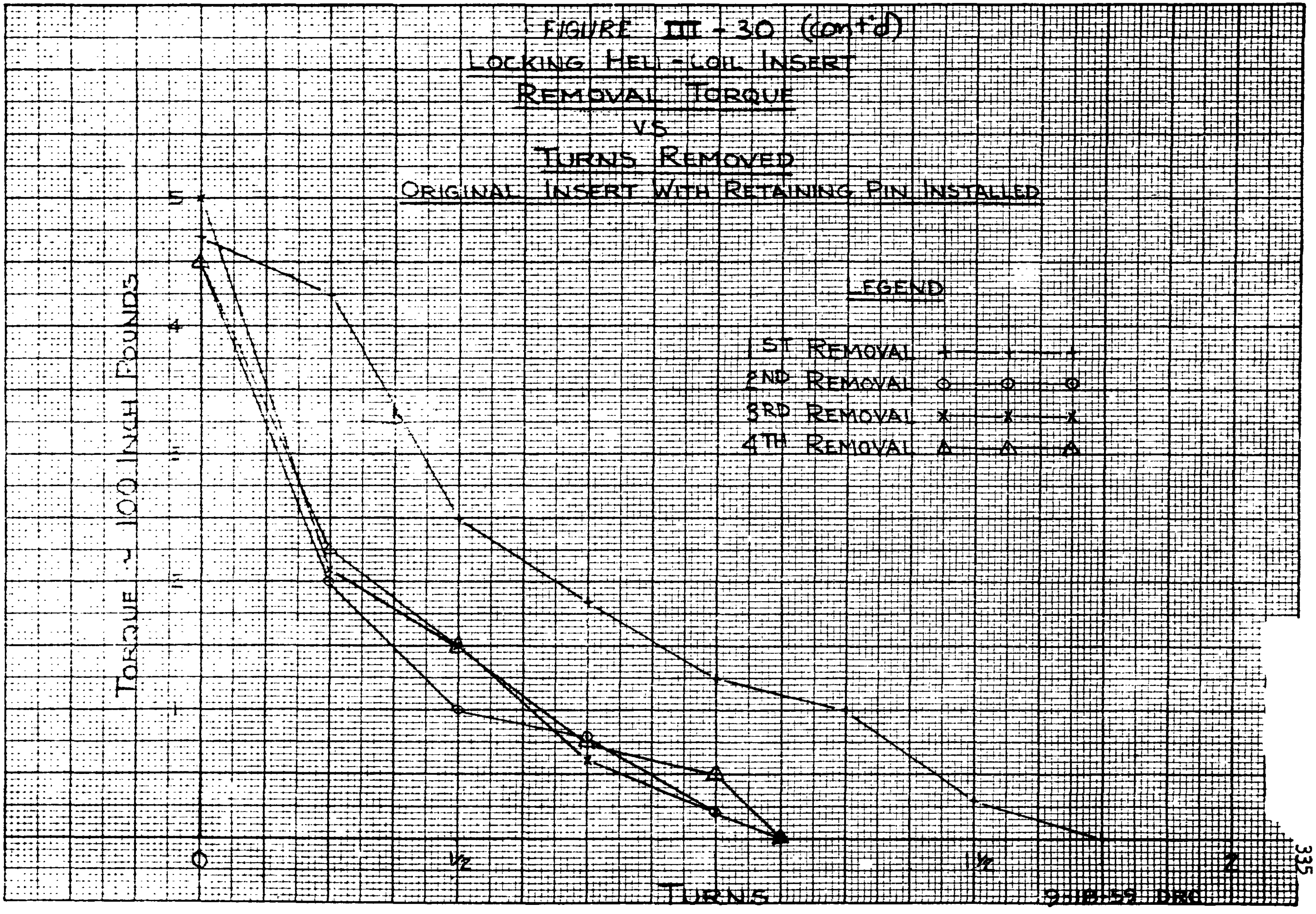


FIGURE III - 30 (cont'd)
LOCKING HELI-COIL INSERT
REMOVAL TORQUE
VS
TURNS REMOVED
ORIGINAL INSERT WITH RETAINING PIN INSTALLED
LEGEND
1ST REMOVAL
2ND REMOVAL
3RD REMOVAL
4TH REMOVAL
TORQUE ~ 100 INCH POUNDS
5
4
0
1/2
1 1/2
2
TURNS
9-18-59 DRG

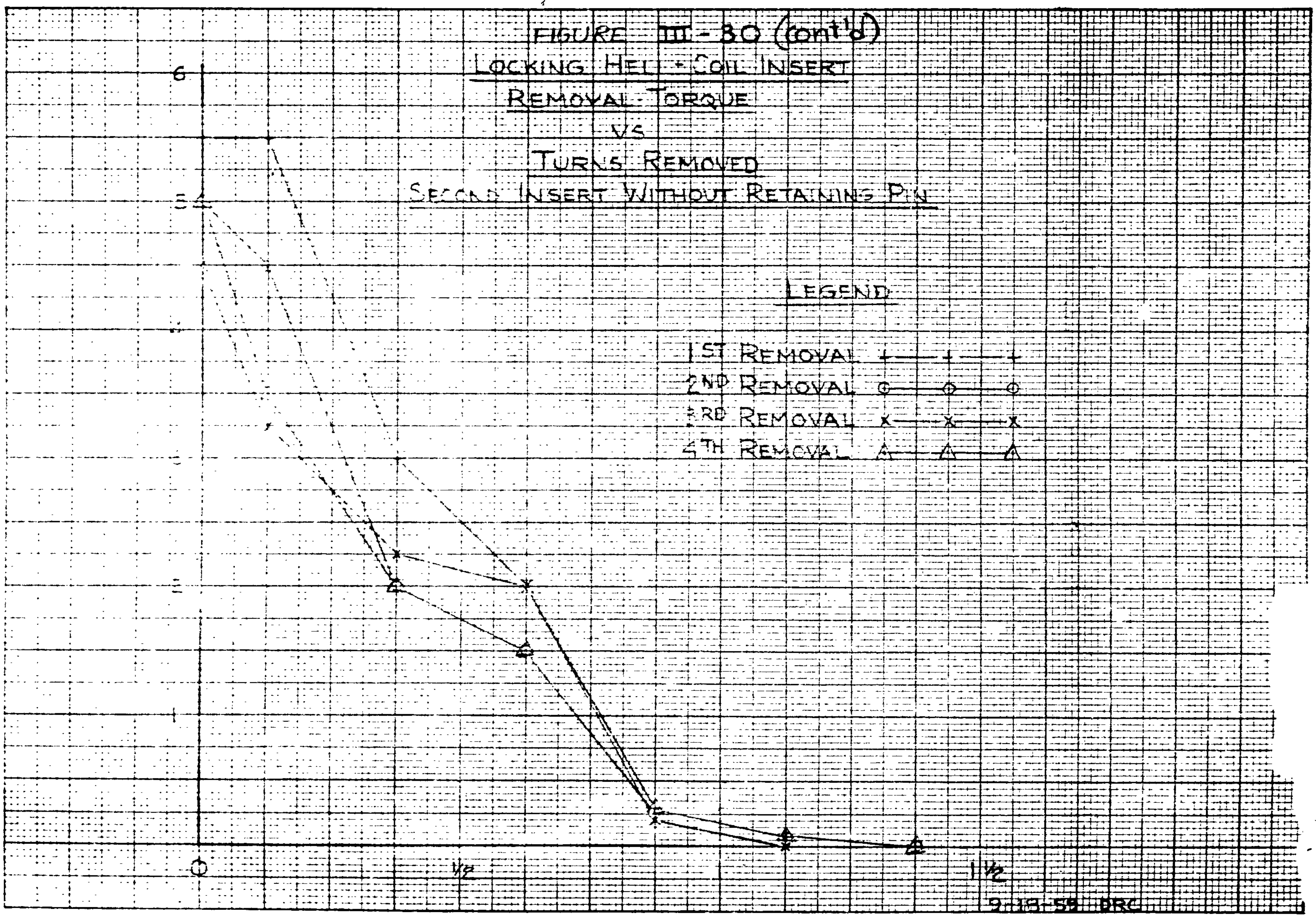


FIGURE III - 30 (cont'd)
LOCKING HELI - COIL INSERT
REMOVAL TORQUE
VS
TURNS REMOVED
SECOND INSERT WITHOUT RETAINING PIN
LEGEND
1ST REMOVAL
2ND REMOVAL
3RD REMOVAL
4TH REMOVAL
6
0
1/2
1 1/2
9-18-59 DRC

for 40,000 cycles, minus 65°F for 10,000 cycles, and 450°F for 1,000 cycles. The total of 51,000 cycles was completed without leakage. The plug was removed and torque readings were taken at each quarter turn during removal. The insert had backed out of the manifold during impulse cycling or during plug removal. The plug was installed and removed seven additional times with torque recordings taken as before. There was no additional movement of the insert in the manifold. The decay of the torque required for installation and removal of the plug with respect to the number of installations is presented in Figure III-31. The first installation and removal of plug, S/N 2, and manifold, S/N 2, resulted in the insert backing out of the manifold completely. A new insert was installed and pinned in the manifold. The pin was a portion of a drill which was broken off during the drilling operation. The insert again unscrewed from the manifold during the first removal of the plug. Galling of the threads in both installations caused high torques which resulted in the binding of the insert to the plug.

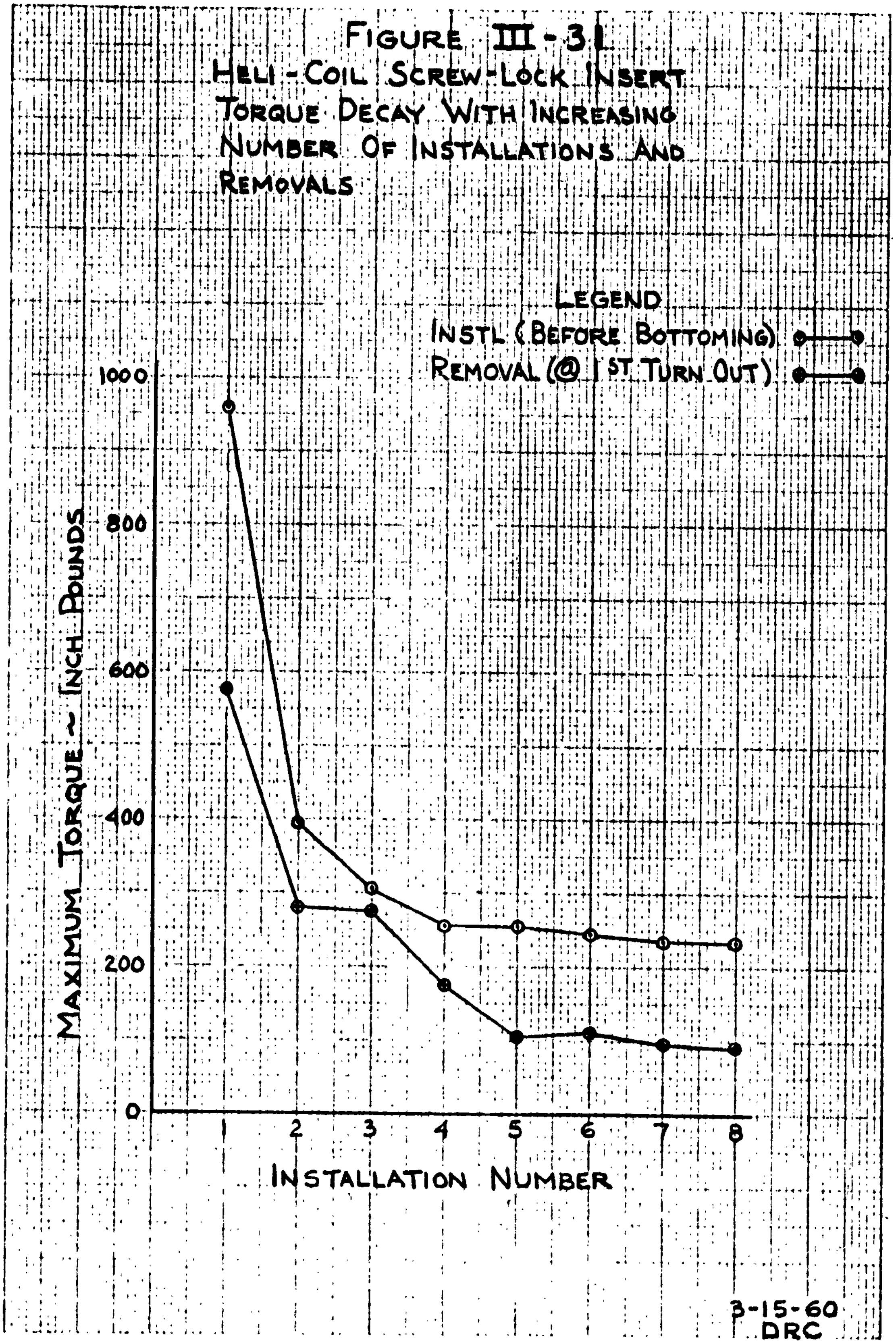


FIGURE III-31
HELI-COIL SCREW-LOCK INSERT
TORQUE DECAY WITH INCREASING
NUMBER OF INSTALLATIONS AND
REMOVALS
LEGEND
INSTL (BEFORE BOTTOMING)
REMOVAL (@ 1ST TURN OUT)
MAXIMUM TORQUE ~ INCH POUNDS
1000
800
600
400
200
0
1
2
3
4
5
6
7
8
INSTALLATION NUMBER
3-15-60
DRC

LONG LOK TEST

A sample of a Long Lok was obtained for evaluation for possible use on the hydraulic modules installed in manifolds. The locks consist of a small section of stainless steel tubing pressed into a precut slot in the threaded portion of the module as shown in Figure III-32.

A test block with a threaded hole to match the module was fabricated for the test. The block was heat treated to the approximate Rc hardness that will be used in the actual manifolds.

The test sample, a 10-32 bolt, was inserted in the test block and finger tightened until the lock was engaged. The test sample was then tightened with a torque wrench and the torque noted. The torque required to tighten the bolt one revolution varied as the bolt rotated. (See Figure III-33) Figure III-34 presents the torque required to insert and remove the test bolt the first time it was installed in the test block. Figure III-35 presents the same data for the second installation of the test bolt in the test block.

1. The torque variation during each turn, as shown in Figure III-33 indicates that the lock is not as effective after a thread groove is formed in the lock tube.

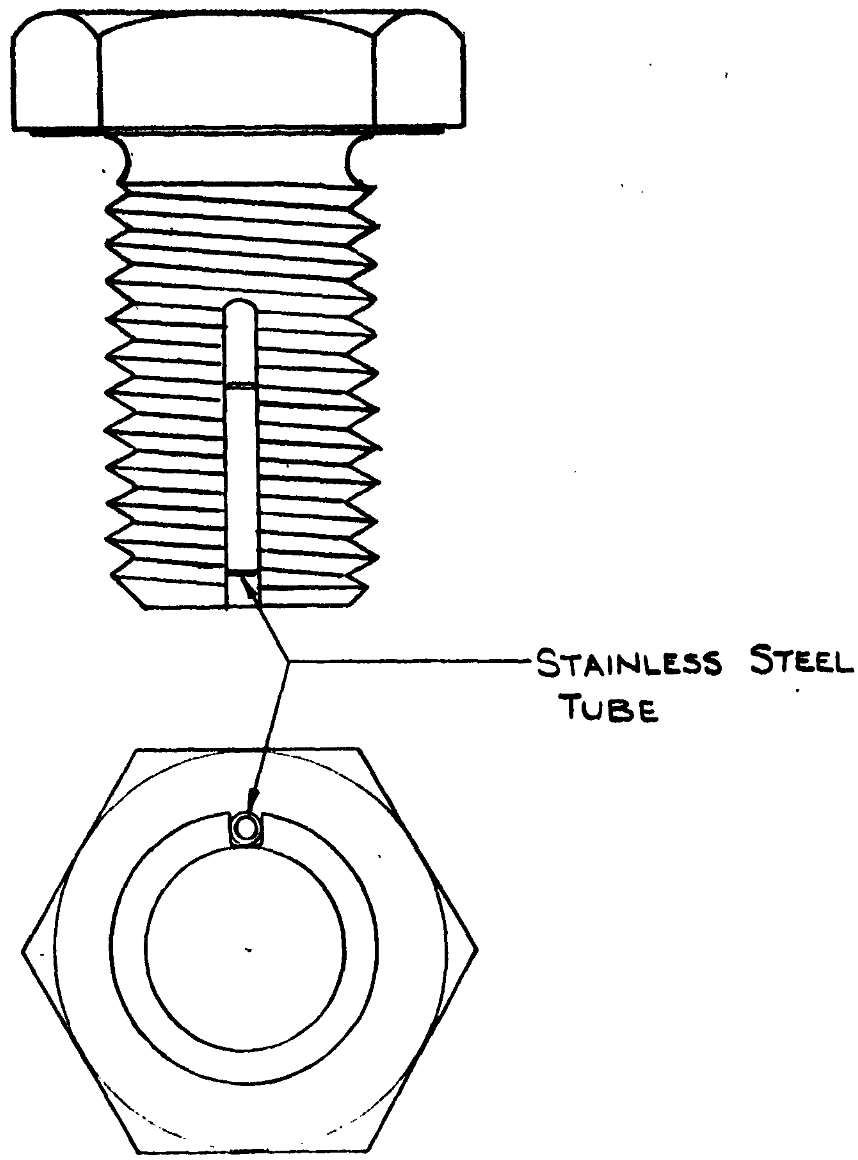
2. The torque required to remove the bolt was substantially less than that required to insert the bolt on the first installation, as shown in Figure III-34.

3. The torque required to insert and remove the bolt on the second installation is less than that required on the first installation. This reduction can be observed by comparing Figures III-34 and III-35.

A test block to accept a 3/4 inch bolt incorporating a "Long-Lok" thread lock was fabricated from 4140 steel having a hardness of Rc 41 (188,000 psi). Two bolts with "Long-Lok" thread locks were installed in and removed from the test block twice. The torque required to turn the bolt was recorded at every 1/8th of a turn. The results of these tests are shown in Figures III-36 through III-39. The torque required to remove the bolts after the first installation was much less than the torque required to install the bolt. The reduction of the torque on the second installation indicates the re-use of the "Long-Lok" to be impractical.

FIGURE III-32

LONG LOK



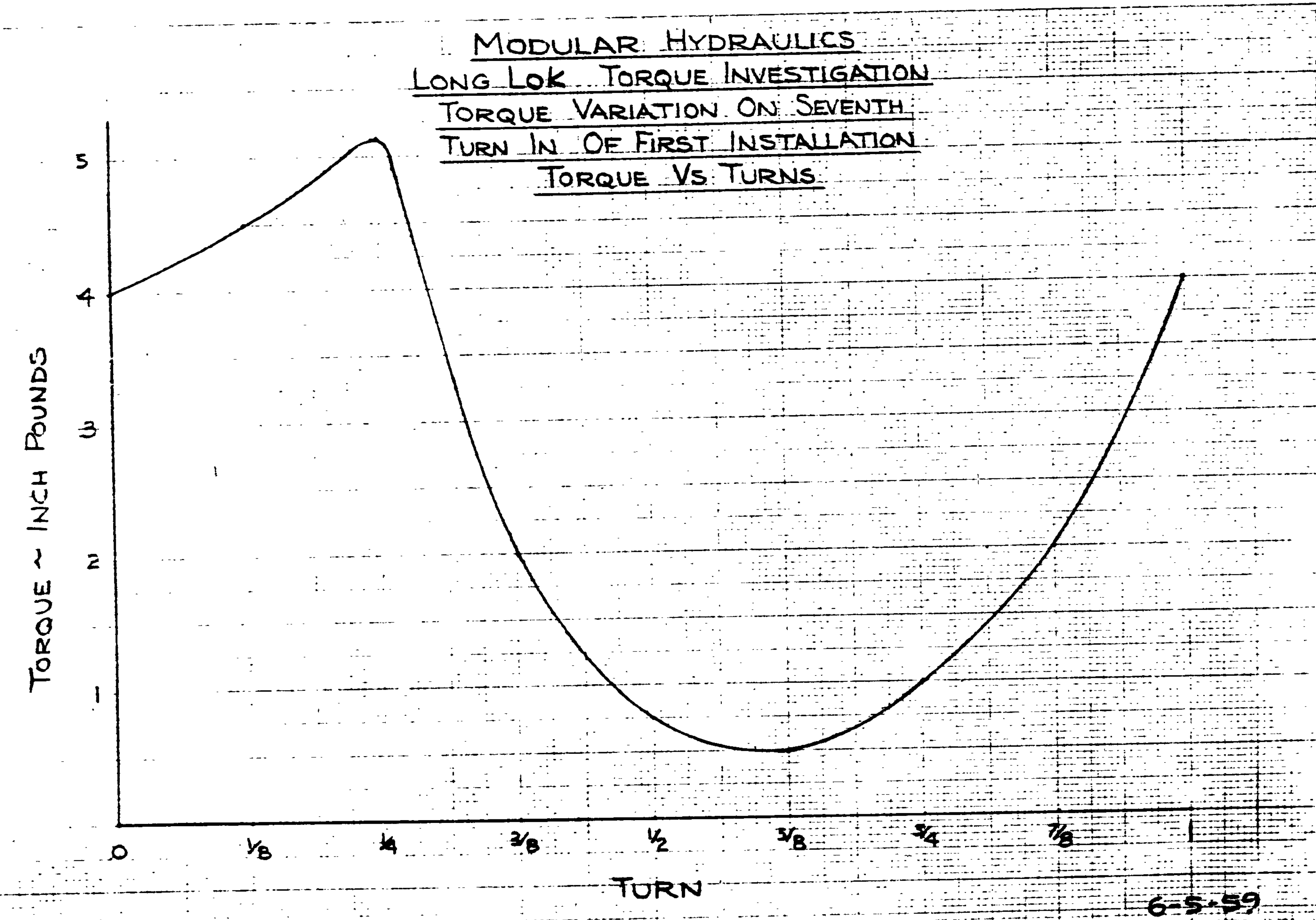


FIGURE III-33

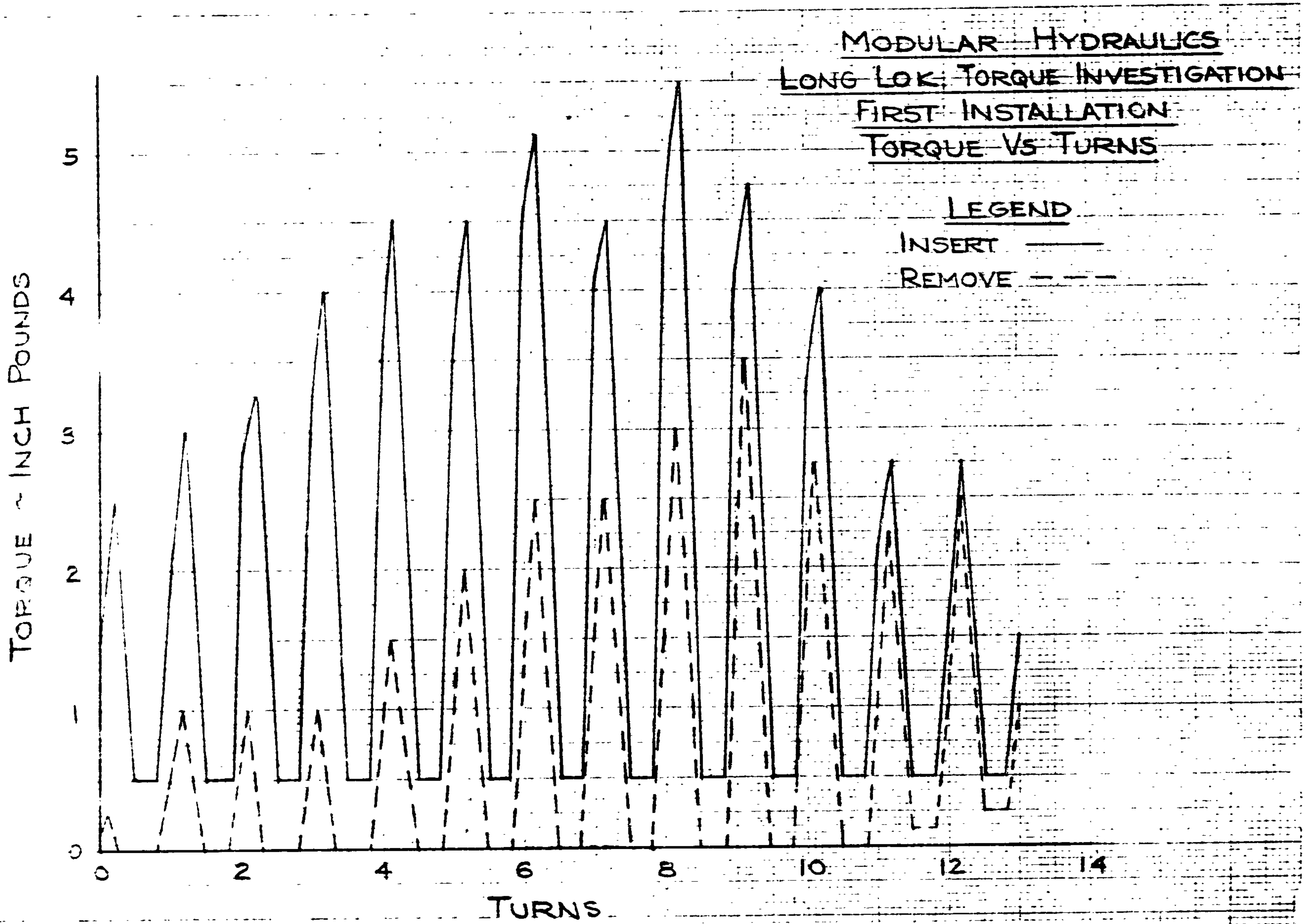


FIGURE III-34

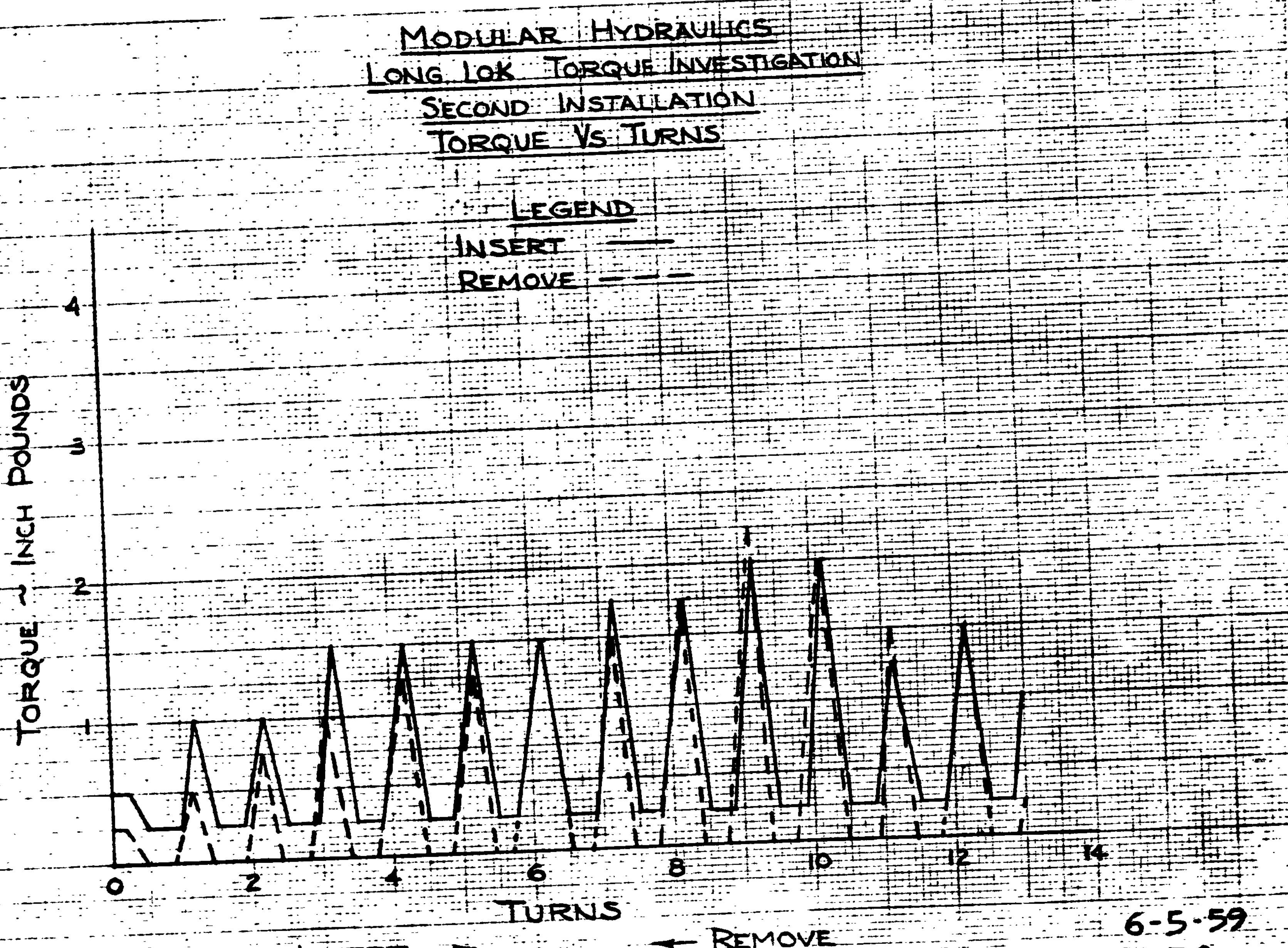


FIGURE III-35

FIGURE III-36

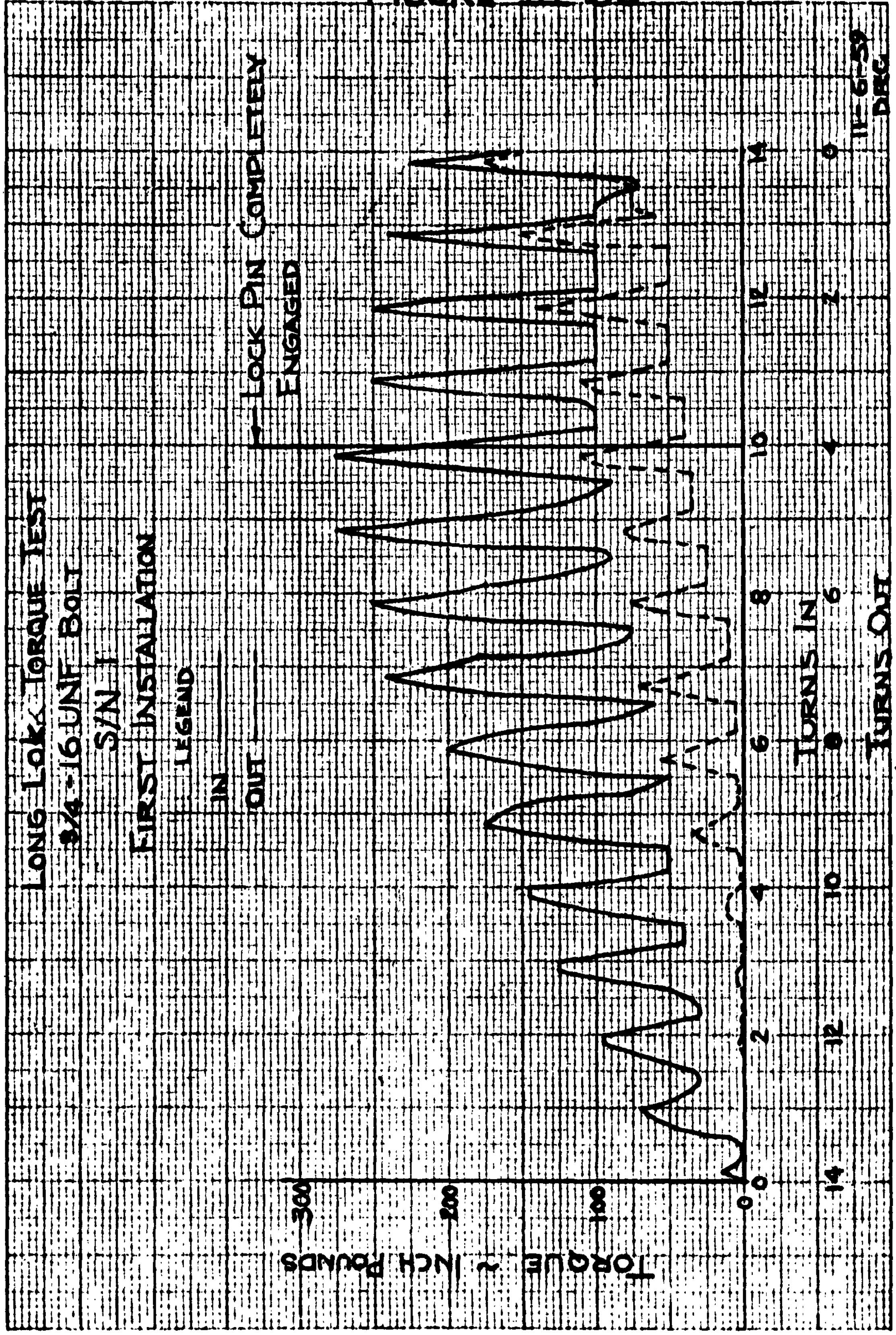


FIGURE III-37

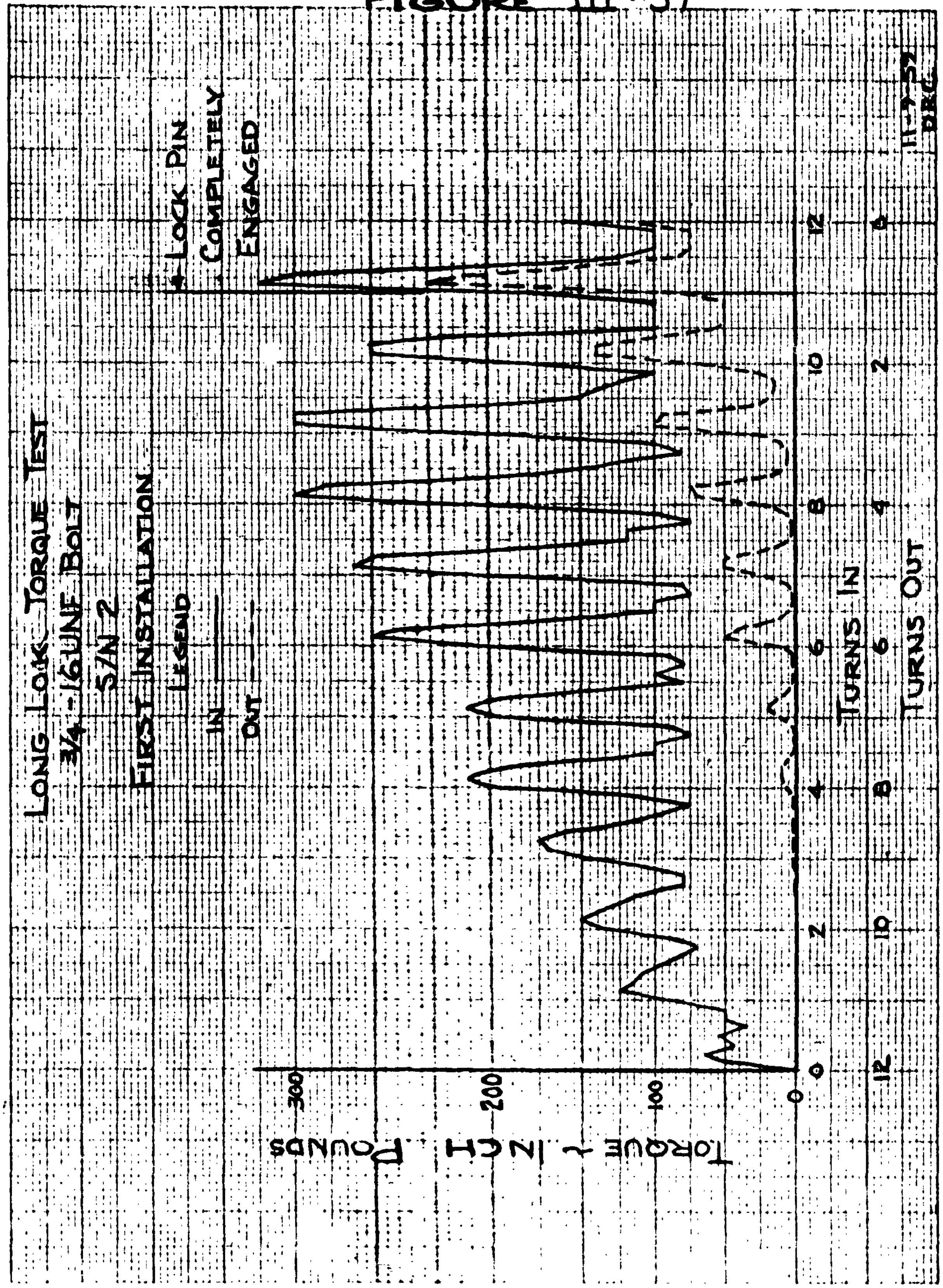


FIGURE III-38

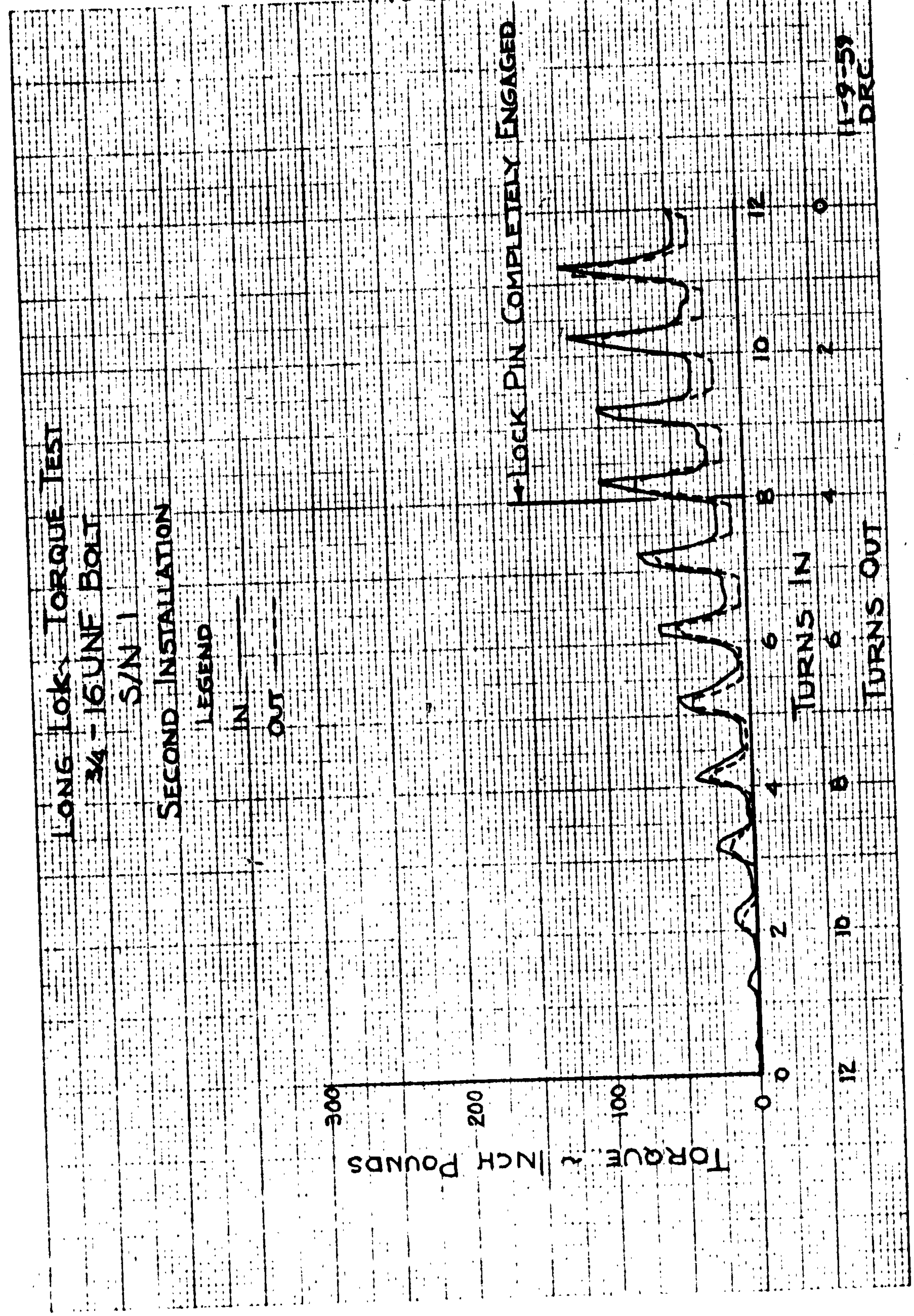
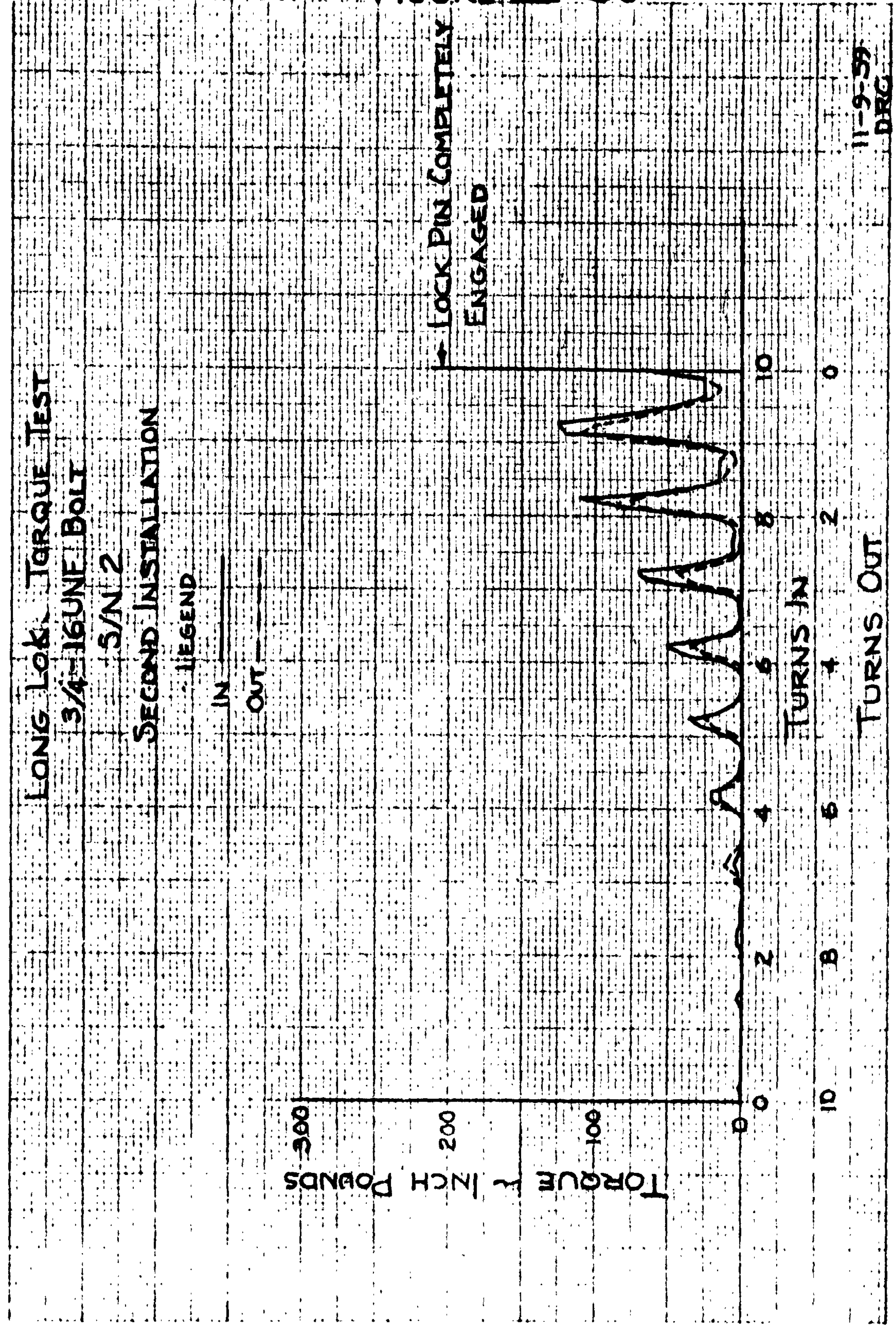


FIGURE III-39



APPENDIX III-5

PACKAGE TEST RESULTS

PURPOSE - The package tests were conducted simultaneously with the component development and qualification which was a part of the Phase II effort. These tests were designed to answer the questions: (1) Do the components, when grouped, have any interaction upon each other which will cause a given component to perform differently than it does when tested singly? (2) Are the components satisfactory as they are being qualified or should design modifications be made? (3) What is the magnitude of pressure loss which will occur in the intricate passages of a typical package? (4) Will any structural weaknesses show up in the packages because of asymmetrical design and a resulting unequal distribution of stresses? The discussion which follows will show that the only structural problems which were met consisted of part deflection under pressure which allowed leakage from seal installations. Face mounted seal installations must be made very rigid because of the low spring back inherent in a metallic seal. Several design inadequacies showed up in the various components. These were corrected by working with the manufacturers responsible for the development of the units. The pressure drop tests are inconclusive. More work should be done in this area, inasmuch as the measured losses bear no agreement with the calculated losses for some passage configurations.

NUMBER 1 PACKAGE TESTS - The No. 1 Package is a manifold containing three modular components. These components are: (1) a three way - two position solenoid operated selector valve, (2) a two way restrictor and (3) a check valve. These components had satisfactorily completed the evaluation test prior to installation in the package. The check valve operated satisfactorily throughout the package test program but difficulties were

experienced with the selector valve and two-way restrictor. The selector valve hung in the open position during high temperature operation and was returned to the manufacturer for rework. After this rework the valve operated satisfactorily during a complete re-run of the evaluation test and the remainder of the package test program. The two way restrictor filter screen in the six gpm flow path came loose and was found in the selector valve cavity of the manifold. The manufacturer of this restrictor also encountered this difficulty in development tests and has eliminated it by adding baffles within the restrictor.

The results of the package test program were satisfactory with the exception of the pressure drop data. The value of the pressure drop data with the components replaced by test plugs is questionable due to the large differences in the data recorded for the two flow paths tested. The dynamic effects of the fluid stream appear to provide a restrictive action in one case and little or none in the other. If this is true, the cavity configuration; size, shape and porting, and direction of flow influence the pressure drop to such an extent as to render the data useless as a basis for comparison to pressure drop data taken with the components installed.

NO. 1 PACKAGE
PRESSURE DROP DATA

Test Para.	Flow GPM	G_1	G_2	G_3	G_4	P	Remarks
4.4.1	2		900	120		780	
	3		970	140		830	
	4		1180	160		1020	
4.4.2	2	420	80			340	
	3	660	100			560	
	4	1040	160			880	
4.4.3	2	240	80			160	Components replaced by Test Plugs
	3	320	120			200	
	4	390	170			220	
4.4.4	2		60	80		-20	Components replaced by Test Plugs
	3		80	100		-20	
	4		100	100		0	

NUMBER 2 PACKAGE TESTS - The No. 2 Package is an actuator with a manifold in the lug end-cap. The manifold provides cavities for three 4 gpm components. These components, which are a one way restrictor, a shuttle valve, and a relief valve, had satisfactorily met the evaluation test requirements prior to installation in the test package. The only difficulty experienced with the components was sticking of the relief valve during low temperature operation. The valve was adjusted to relieve at a differential pressure of 4800 psi at room temperature, but when the system temperature was lowered to minus 15 degrees F the valve operated only once. The pressure was increased to 5200 psi on the second pressure application without opening the valve. The valve was heated to room temperature, and operational checks showed the valve would operate satisfactorily. The temperature was lowered to minus 10 degrees F and a pressure of 5100 psi was applied without opening the valve. The valve adjustment was reset to open at a differential pressure of 4600 psi at a temperature of 40 degrees F. The temperature was reduced to minus 15 degrees F and the valve operated at a differential pressure of 4800 psi. The valve failed to open on the second pressure application, but operated properly on subsequent pressure applications. The manufacturer did not experience any difficulties of this type during qualification tests of the valve. The failure to open could possibly be attributed to difference in the ambient and oil temperatures. The ambient temperature was minus 80 degrees F while the oil temperature was 10 degrees F. This temperature difference and the heat generated at the valve poppet might cause binding of the poppet.

The seal between the actuator barrel and end cap manifold leaked when pressure was applied to the actuator. The seal squeeze was increased

to the design maximum by machining material from the face of the manifold flange. The joint still seeped when subjected to 5000 psi. This seepage appeared uniformly around the joint between the bolts. The seepage apparently results from separation of the joint with pressure application. No further effort was made to correct the seepage because the system pressure of 4000 psi caused no seepage.

The pressure drop data obtained from this package is more consistent with calculated losses than that of the Number 1 Package. The test procedure used was the same as that for the Number 1 Package, therefore, the consistence of the data is due to configuration and arrangement of the cavities.

NO. 2 PACKAGE
PRESSURE DROP DATA

Test Para.	Flow GPM	G_1	G_2	Δ P	Temp °F	Remarks
4.1.6	1	100	730	630	15	Δ P Includes drop across selector
	2	120	1160	1040		
	3	140	1440	1300		
	4	200	2330	2130		
4.1.7	1	660	210	450	15	Free flow direction Shuttle Manifold Restrictor
	2	860	160	700		
	3	1130	260	870		
	4	1480	300	1180		
4.1.9 (4.1.6)	1	-	-	-	90	
	2	140	600	460		
	3	150	1120	970		
	4	180	1900	1720		
(4.1.7)	1	-	-	-	90	
	2	400	200	200		Free Flow
	3	520	200	320		
	4	680	210	470		
(4.1.6)	1	-	-	-	450	
	2	110	550	440		
	3	110	860	750		
	4	120	1550	1430		
(4.1.7)	1	-	-	-	450	
	2	300	180	120		Free Flow
	3	350	190	160		
	4	440	200	240		
4.2.1	2	100	4000	3900	90	Plug installed in place of Restrictor.
	3	120	4300	4180		
	4	150	4400	4250		
4.2.2	2	110	230	120	90	Test Cap installed in place of Relief Valve
	3	110	300	190		
	4	140	440	300		
4.2.3	2	110	320	210	90	Test Cap installed in place of Restrictor
	3	120	480	360		
	4	150	590	440		
4.2.4	2	400	180	220	90	Same as 4.2.3 (Flow Reversed)
	3	500	200	300		
	4	630	220	410		

NO. 2 PACKAGE
PRESSURE DROP DATA

(CONTINUED)

Test Para.	Flow GPM	G_1	G_2	G_3	P	Temp °F	
4.2.5	2 3 4		190 210 220	310 450 550	120 240 330	90	

NUMBER 5 PACKAGE TESTS - The Number 5 Package is a two part manifold which has five components in one part, the required porting in the other part and automatic shut-off devices in the four passages connecting the two parts. The five components are: (1) a pressure switch, (2) a class "C" relief valve, (3) a solenoid operated shut-off valve, (4) a class "B" filter, and (5) a class "C" filter. The component function and leakage tests were completed satisfactorily in accord with the test procedure with the exception of the relief valve and filter tests. The relief valve full flow pressure was adjusted from the required 4850 psi differential pressure to 4050 psi differential pressure. This adjustment was necessary in order to eliminate the pressure oscillations which are present at the full flow condition of the relief valve at the original setting. Upon disassembly and inspection it was found that this valve was not made per blueprint.

Both the class "B" and "C" filter internal shut-off valves which prevent the system draining when the filter bowl is removed, leaked excessively. These filters were fabricated from development castings which distort under pressure allowing leakage at the shut-off valves. The filters were reworked to replace the original castings with redesigned castings which eliminate the distortion and in turn the leakage at the shut-off valve. The filter element of the class "B" filter was found to be hung in the filter bowl when the dummy element was to be installed for the element differential pressure indicator tests. The element could not be removed by hand, but was removed without damage by gripping the element end in a vise and tapping the bowl with a hammer. The filter was assembled and the element differential pressure indicator test

conducted. The indicator of the class "C" filter would not reset after the third actuation. The filter bowl was rapped with a hammer and then the indicator reset properly.

The leakage tests of the automatic shut-off valves between the parts of the manifold resulted in excessive leakage at two of the four valves tested. The two leaking poppets were lapped to their seats with some success, but leakage would still occur if the poppets were caused to be moved on the seat. The poppets were not guided sufficiently due to the clearance between the poppet skirt and seat barrel to insure proper seating of the poppet. The pressure drop data for this package was satisfactory. The peculiarities exhibited by the Number 1 Package are not present.

The Hi-Ceals between the two parts of the manifold leaked and the replacement of these seals with elastomer o-rings was necessary to conduct the test program. The leakage by the metal seals could be reduced by increasing the torque on the bolts joining the two parts of the manifold but leakage occurred at 2000 psi at the maximum torque valves of the bolts (1/4 inch-400 inch pounds, 3/8 inch-910 inch pounds, 7/16 inch-1200 inch pounds). After the completion of the tests, the manifold parts were separated and the seal seats were polished. The manifold was assembled using metal seals. The bolts were tightened to torque valves of 200, 455 and 600 inch pounds, respectively, for 1/4, 3/8 and 7/16 inch diameter bolts. Leakage occurred at the parting plane of the manifold at 1200 psi. The torque on the bolts was doubled to attain the maximum torque valves of the bolts but leakage occurred when a pressure of 2000 psi was applied.

NO. 5 PACKAGE
PRESSURE DROP DATA

Test Para.	Flow GPM	G_1	G_2	G_3	G_4	ΔP	$\Delta P'$	Remarks
4.1.1	5	200	140			60		
	10	220	170			50		
	15	330	260			70		
	20	460	360			100		
	25	650	500			150		
4.1.2	5	150	130			20	-40	$\Delta P'$ thru
	10	220	190			30	-20	CVP-3353-3
	15	320	290			30	-40	Filter Assy
	20	440	400			40	-60	
	25	630	530			100	-50	
4.1.3	3	158			87	71		
	6	238			106	132		
	9	340			140	200		
	12	470			192	278		
4.1.4	3	175			108	67	+4	$\Delta P'$ thru
	6	245			128	117	-15	CVP-3356
	9	345			162	183	-17	Sol. Oper Shut-off
	12	460			214	246	-32	Valve
4.1.5	3	115			103	12	-55	$\Delta P'$ thru
	6	150			124	26	-91	CVP-3353-2
	9	204			160	44	-139	Filter Assy
	12	295			210	85	-161	
4.1.6	5	2900		130		2870		
	10	3000		180		2820		
	15	3040		290		2750		
	20	3080		430		2650		
	25	3700		590		3110		
4.1.7	5	132		108		24	2846	$\Delta P'$ thru
	10	210		175		35	2785	CVP-3350-2
	15	330		280		50	2700	Relief Valve
	20	490		415		75	2575	
	25	720		590		130	2980	

NO. 5 PACKAGE
PRESSURE DROP DATA

(Continued)

Test Para.	Flow GPM	G_1	G_2	G_3	G_4	ΔP	$\Delta P'$	Remarks
4.1.8	5 10 15 20 25	160 242 365 500 720	115 162 240 340 490			45 80 125 160 230	-15 +30 +55 +60 +80	
4.1.9	3 6 9 12	178 242 340 468			98 114 145 184	80 128 195 284	+9 +4 -5 -6	ΔP thru Auto-Shut Off (4.1.3)